29489

National Research Council Canada
Conseil national de recherches Canada

Seventh NRCC Symposium on Industrial Application of Gas Turbines

Septième Symposium du CNRC sur les applications industrielles des turbines à gaz

Banff, Alberta
September 1987

Banff, Alberta
septembre 1987

Canada

Proceedings

Seventh NRCC Symposium on Industrial Application of Gas Turbines

Compte rendu

Septième Symposium du CNRC sur les applications industrielles des turbines à gaz

The Banff Centre
Banff, Alberta
27 – 30 September 1987

The Banff Centre
Banff, Alberta
27 –30 septembre 1987

Sponsored by:
National Research Council Canada
Associate Committee on Propulsion
Subcommittee on Industrial
Application of Gas Turbines

Sous l'égide du :
Conseil national de recherches Canada
Comité associé de la propulsion
Sous-comité des industrielles des
turbines à gaz

CONTENT

SUBCOMMITTEE MEMBERSHIP

INTRODUCTION by P. Anderson, Conference Chairman

KEYNOTE ADDRESS

SESSION 1: MAINTENANCE SYSTEMS

Session Chairman: D. Protheroe - Rolls Royce (Canada) Ltd.

PAPER NO.

On Site Heavy Maintenance Techniques 1
J.P. Inglis; K.C. Hobbs

Gas Turbine Maintenance Experience 2
A. Dellora

Development and Application of Diffusion Braze Repair Techniques 3
S.G. Stoykewich; R.J. Armstrong

SESSION 3: IMPACT OF NEW TECHNOLOGY

Session Chairman: L. Tessier - Esso Resources Canada Ltd.

Operation of Oil-less Seal and Bearing System for Centrifugal Compressors 4
R.P. Lancee

Precise Gas Turbine Fuel Control 5
G.E. Smith

Technology Trends in Aircraft Gas Turbines and Their Industrial Applications 6
S.P. Sampath

Emerging Technologies for Life Cycle Management of Turbine Engine Components 7
A.K. Koul; J-P. Immarigeon; W. Wallace

SESSION 3: IMPROVING EXISTING MACHINES

Session Chairman: L. Williams - TransCanada Pipelines Ltd.

Improvements in the Performance of a Regenerative Gas Turbine with Blade Rejuvenation and Modification to the Cooling and Sealing Systems 8
D. Little

Avon Series and Cooper RT.48S Upgrade 9
P.J. Handy; R.H. Meier

New or Replacement Power with the First "Third Generation" Aero-Derived Gas Turbine 10
G.N. Rogers

Conversion of Gas Compressors To Improve Efficiency 11
R.H. Meier

SESSION 4: Performance Testing

Session Chairman: B. Hoyland - Solar Turbines Canada Ltd.

PANEL DISCUSSION

Panelists/Invites

J. Scott John Scott Engineering
NOVA, An Alberta Corporation
Solar Turbines Canada Ltd.
TransCanada Pipelines Ltd.

SESSION 5: CONDITION MONITORING

Session Chairman: H.I.H. Saravanamuttoo - Carleton University

The Development of a Compressor Station Health Monitoring System 12
T. Van Hardeveld

A Critical Assessment of the Application of Artificial Intelligence to Engine Health Monitoring 13
B.D. MacIssac

Predicting the Effects of Progressive Compressor Fouling on Gas Turbine Performance 14
G.F. Aker; H.I.H. Saravanamuttoo

Vibration Monitoring Experience in Gas Turbine Engines of the Canadian Navy 15
P. Cheney

ASSOCIATE COMMITTEE ON PROPULSION
SUBCOMMITTEE ON INDUSTRIAL APPLICATION OF GAS TURBINES

COMITE ASSOCIE DE LA PROPULSION
SOUS-COMITE DES APPLICATIONS INDUSTIELLES DES TURBINES A GAZ

SEPTEMBER 1987 SEPTEMBRE

Mr. P. Anderson, Chairman
NOVA, An Alberta Corporation
P.O. Box 2535
Calgary, Alberta
T2P 2N6

Mr. J. Bird
Engine Laboratory
Division of Mechanical Engineering
National Research Council Canada
Montreal Road
Ottawa, Ontario
K1A 0R6

Mr. P. Cheney
Department of National Defence
National Defence Headquarters
101 Colonel By Drive
Ottawa, Ontario
K1A 0K2

Mr. C.P. Conquergood
Shell Canada Resources Ltd.
Production Division
P.O. Box 100
Calgary, Alberta
T2P 2H5

Prof. R. Evans
Department of Mechanical Engineering
University of British Columbia
2075 Wesbrook Mall
Vancouver, British Columbia
V6T 1W5

Mr. J.-M. Fournier
Pratt and Whitney Canada Ltd.
Industrial Marine Division
P.O. Box 10
Lonqueuil, Quebec
J4K 4X9

Mr. B. Hoyland
Solar Turbines Canada Ltd/Ltee
2510 - 84th Avenue
P.O. Box 5630
Edmonton, Alberta
T6C 4G1

Mr. D.C. Hubble
Ontario Hydro
700 University Avenue
Toronto, Ontario
M5G 1X6

Mr. B. Junkin
Standard Aero Ltd.
Winnipeg International Airport
33 Allen Dyne Road
Winnipeg, Manitoba
K3H 1A1

Mr. R. Morland
Esso Resources Canada Ltd.
Judy Creek Area Operations
Mail Bag 1000
Swan Hill, Alberta
T0G 2C0

Mr. D. Prothereo
Rolls Royce (Canada) Limitee
P.O. Box 1000, Montreal-AMF
Montreal, Quebec
H4Y 1B7

Prof. H.I.H. Saravanamuttoo
Department of Mechanical and
Aeronautics Engineering
Carleton University
Ottawa, Ontario
K1A 5B6

Mr. L. Williams
TransCanada Pipeline Ltd.
P.O. Box 54
Commerce Court West
Toronto, Ontario
M5L 1C2

Dr. I.G. Campbell
Gas Dynamics Laboratory
Division of Mechanical
Engineering
National Research Council
Canada
Ottawa, Ontario
K1A 0R6

ON SITE
HEAVY MAINTENANCE TECHNIQUES

BY

J.P. INGLIS VICE PRESIDENT, PRODUCT SUPPORT

K.C. HOBBS DIRECTOR, PRODUCT SUPPORT

SEPTEMBER 1987

ON SITE MAINTENANCE TECHNIQUES

And in the beginning was the Avon. And the Avons went out into the wilderness, multiplied, became a multitude, and begat problems. And the problems themselves multiplied, but because of improved communications, it did not take 40 days nor 40 nights for the messages to get back to base. As North American operators are vocal and explicit with it the theme was clear and distinct. "Arise, get off your bed, and come and see at first hand." The Company accepted the challenge and I was despatched out into the wilderness North Americans call a kitchen and was astonished at the heat (or cold) therein.

Whereas in most aero engine operations when things go wrong, engines are usually returned to an overhaul base or to the manufacturers for rectification, it soon became clear that using the same philosophy with industrials would result in high transportation costs and long times out of service simply because many pipeline stations are many hundreds of miles away from the nearest overhaul base. It was decided, therefore, to look at the individual problems that were besetting the operation and to invent tooling that would enable us and or our Operators to change unit assemblies in the field. This tooling became known as "Heavy Maintenance Tooling".

Cold End Problems

Initially the Avon suffered from front end problems, affecting both rotating and static parts. Looking at each major assembly in turn, we found we had a number of problems in the front bearing housing assembly, some in the front compressor stator assemblies and some concerning the front row of compressor rotors.

Front Bearing Housing Problems

There were two problems here, cracking of the intake guide vane skins and bush wear.

Intake guide vane (IGV) cracking was found to be due to high standing stresses in the skin (due to anti-icing air) combined with high alternating stresses due to rotor passing frequencies, the latter magnified by the extra high mass flows encountered during the intense cold of winter operation. Two modifications came out of this investigation, firstly thickened skin vanes and secondly reduced pressure anti-icing air.

Front Bearing Housing Problems (continued)

Changing guide vanes in the field could only be done by changing front bearing housing assemblies achieved by first removing the starter drive mechanism as shown, and then by removing the front bearing housing (Fig. 1 & 2). Fitting the new front bearing housing was the reverse of this.

IGV bush wear was again a combined effect problem being caused primary by frettage exacerbated by dirt and solid particulate matter ingested with the air. The cure was to exchange the chrome plate on the IGV journals for tungsten carbide and to incorporate the so treated IGV's by exchange of FBH assembly as explained earlier.

Compressor Casing and Stator Problems

The only significant problem encountered with the stator blades has been failure of the stage 1 vanes (third stage). The failure problem was reduced fatigue strength due to corrosion attack combining with relatively minor resonances. Coating the blade all over with PTFE guarded against corrosion while detuning by rivetting the vanes together into quadrants and dowelling the quadrants together got rid of the resonances.
As there was an associated frettage problem, this was dealt with by making the blades circumferentially an interference within the compressor casings. Thus the casings and stators could only be changed as a matched pair. This was achieved by transferring the GG from horizontal to vertical position (see Fig 3 & 4) and by removing and replacing the casings as per Fig 5.

Later in the operation we had a few casings themselves fail by a corrosion mechanism originating in the seventh stage bleed valve opening. The cure was to fit heavy duty aluminium casings in place of the original lightweight magnesium casings, by the same method as outlined above.

Compressor Rotor Problems

As introduced into industrial service the Avon was soft nosed having aluminium rotors for the first ten stages, 00 through 8. Modifications became progressively available that changed stages 3 through 8 to titanium blading, and later the 00 stage also, but these were most conveniently changed out at overhaul. However, there were a number of problems associated with the aluminium 00 rotors which were successfully dealt with in the field.

00 Stage Rotor Problems

1) Ice Ingestion

Whereas there was not a problem with the engine's own protection system, which was perfectly adequate, sometimes the intake houses leaked, and ice which subsequently formed inside would break off and be ingested as large lumps. These lumps permanently distorted the aerofoils which tried to digest them in varying degrees. Where distortion had not been severe enough to cause the rotors to strike the rear of the IGV's, the procedure was to remove the FBH as already described and then to pull the 00 wheel off the front of the compressor shaft as per Fig 6. The distorted blade was then replaced by another of the same weight and the gas generator (GG) rebuilt.

2) Cracked 00 Rotor Lugs

This occurred on aluminium rotors, due to a frettage corrosion mechanism. Cracks could be detected before progressing to failure using an eddy current detector during routine inspection. The subject blades could then be changed out as above.

3) Cracked Aerofoils

This occurred on peak loppers situated in or near the heart of industrial cities where atmospheric pollution was so bad that gooey deposits up to 0.1" thick would build up all over the aerofoil. The deposits reduced the aerofoil first pin rolling frequency by some ten percent, thereby bringing the first major resonance within the engine speed range. Frequent hand cleaning and local dye penetrant crack testing was successful in containing the situation. Cracked blades were individually changed out as required.

All of these problems resulted in the introduction of 00 rotors in titanium which is much stiffer and stronger than aluminium and has excellent corrosion resistance. The heavier titanium blades necessitated a heavier wheel and pins and even although the rotor assemblies were interchangeable we were never able to upgrade to titanium in the field due to vibration problems.

Central Engine

The center part of the engine comprising the rear six stages of the compressor within the outlet casing, the internal wheelcase and central thrust bearing never gave us any problems, so no heavy maintenance techniques were required to deal with this area.

Hot End

Flame Tube Location Wear

The biggest single problem to hit the Avon by far, was flame tube location wear. Continuous operation at sea level, particularly during the cold winter months in Canada and North America when the mass flow is increased by 30% or more, resulted in much higher loadings and buffetting of the suspension than we expected.

The first step to introduce a modified radial support channel with central stiffener, and controlled inward loading of each tube during build. All mating faces were hard chrome plated all over. The chrome was later replaced by tungsten carbide and the tubes bolted together after assembly. This standard was an immense step forward and could be changed in the field or local workshop by first removing the low pressure turbine (Fig 7) and then changing the flame tubes as per Fig 8.

Unfortunately the weak link in this arrangement turned out to be the bolting, forcing us to go with flame tube assemblies welded together by the so called Crook's washers. These washers have now been replaced by bridge pieces which are much more easily removed and rewelded in conjunction with the thickened and strengthened back end tubes. Thus the way is clear for any Operator who wishes to upgrade his tubes which includes going to triple dish standard, to do so in the field.

Internal Oil Leaks

The only other significant problem that has beset the hot end, was premature deterioration of the 'O' rings on the tubes which take oil to and from the turbine roller bearing. It was found that by taking the flame tube tooling one more step i.e. designing tooling to remove the HP turbine rotor and nozzle box assembly, that the oil tubes could be easily removed and the 'O' rings replaced. This is shown in Fig 9.

And so the Avon returned from the wilderness, achieved international fame and multiplied even further.

RB211

When designing the RB211 it was decided to make as much use as possible of the fact that this design was totally new, so that we incorporated features which allowed easy interchangeability of components and easy accessibility for inspecting the internals of the engine without the need for excessive strip.

The industrial gas generator (Fig. 10) therefore, is constructed from five separate major sections or modules, any one of which can be removed and replaced without disturbing the component parts of the other sections of the gas generator. This feature allows any one of the modules to be replaced on site or at the operator's own local work shop thereby avoiding the need for the removal of the complete gas generator to an overhaul base. Modules overhauled or supplied new are prebalanced and fully interchangeable. Module interconnections are made either through curvic couplings or splines or at journal bearing locations, thus eliminating the need for gas generator re-balancing after a module has been replaced. Testing after module change is normally limited to a run up to load on site. The modules are of compact dimensions and moderate weight and any necessary replacements can be carried out by the operators and without recourse to heavy lifting equipment. Each module has its own log book in which is recorded the modification standard, component serial numbers and life history.

The major advantages of modular construction are:-

High Gas Generator Availability

The ability to change major sections of the gas generator in the field reduces to an absolute minimum the length of time an installation is inoperative due to reasons of gas generator maintenance or repair and self sufficiency in this respect by the operator can lead to maximum flexibility and economy of operation.

Repair by Local Labor

After removal of the gas generator from its installation, any module can be removed and replaced on site. Module change techniques are not complex and suitable training will permit utilization of local labor. Rolls-Royce offers training facilities in this area.

Ease of Transportation

All modules and tools are transported by truck or helicopter thus easing transportation problems to sites where access is difficult.

Reduced Spares Holding

The need to hold complete spare gas generators is greatly reduced. Spares holding can, with operating experience, be restricted to specific modules with significant savings in capital costs.

Optimised Use of Capital Equipment

The life capability of each module can be fully realised. Expensive repair facilities, tools and equipment are not required by an operator who can easily return a module for repair or overhaul while maintaining operation by use of spare modules only and finally we increase the updating and uprating capability. Engineering improvements which affect both component life and gas generator power output capability can be introduced to customer's gas generators by the change of module.

Identification of the five major modules which together form the basic gas generator, (Fig 11) is given by the reference numbers 01 to 05. All modules are interchangeable and are prebalanced, thus eliminating the need for gas generator rebalancing after a module change. The modules are transported to site in cases of composite metal and fibreglass construction, giving maximum protection without incurring an unacceptable weight penalty especially in those situations where air lifting is required. The cases have provision for both fork lifting and slinging and incorporate a shock absorbant base which forms the main member of the integral transportation stand. Modules are additionally protected by enclosure in a moisture and vapor proof envelope and in the case of rotating assemblies by the provision of special immobilizing equipment. All spare modules are provided with a module change parts kit comprising those consumable items necessary to carry out module replacement. These kits consist of such items as bolts, washers, seals, gaskets, pins and so on.

Taking each module in turn then:-

Module 01 - The Air Intake Casing (Fig 12)

This module comprises the air intake case, variable inlet guide vanes, the actuating ring hydraulic rams, vowles and manifold. The intermediate pressure compressor front bearing, and intermediate pressure spool speed indicating system, the anti-icing hot air manifold and the nose bullet. The casing also carries the attachment point for the gas generator front mounting support strut.

Module 02 - The Intermediate Pressure Compressor(Fig 13)

So called because in the aero engine this is the second stage of compression, the first stage being the large fan. This module comprises the intermediate pressure compressor stator system, that is the half casings and vanes. The intermediate pressure compressor rotor system, discs, rotor and rotor blading and the intermediate pressure compressor rotor front stub shaft. The outer casing is split horizontally giving access to all rotor blades and vanes.

Module 03 - The Intermediate Casing (Fig 14)

Comprises the intermediate case, the internal gear box, starter drive thrust bearing, starter drive shaft and starter drive centrifigal poll mechanism. The high pressure spool speed indicating system, the IP compressor outlet guide vanes, the IP compressor rear stub shaft, the IP compressor location bearing, the HP compressor front stub shaft and the HP compressor location bearing.

Module 04 - The High Pressure System (Fig 15)

This major module combines three sections of the gas generator. They are the high pressure compressor comprising of the high pressure compressor stator system, (i.e. the casings and vanes). The high pressure compressor rotor system, (i.e. discs and rotor blades). The combustion assembly, comprising the HP compressor outlet guide vanes, combustion chamber outer casings, combustion chamber inner casings and the annular combustion chamber and the high pressure turbine comprising the high pressure turbine disc and blades, the bearing stub shaft and the high pressure turbine nozzle guide vanes.

Module 05 - The Intermediate Pressure Turbine (Fig 16)

This module is made up of the IP turbine rotor blades and disc, the IP turbine main shaft, the IP turbine stub shaft, the IP turbine nozzle guide vanes, the high pressure and intermediate pressure turbine bearings, the support assembly for the turbine bearings, and the turbine outer case. The rear flange of this case is the main mounting point for the gas generator.

In addition there are various non modular parts. These are items required for attachment of the modules to each other, nuts, bolts, dowels, washers etc and also those components required for dressing of the core engine so formed.

Included in this group of parts are the air gas starter motor, gas burners, ignitors for the gas burners, a floor mounted fuel manifold, pipes and fittings, the exhaust gas thermocouples, the low tension electrical harness, the IP and HP bleed valves and controls, oil system tubes and fittings, the gas generator front mounting bracket, vibration transducers, pressure transducers, the anti-icing system and the disconnect platform. To follow the procedure established by the use of 01, 02 etc for identification, the non modular parts are grouped in the manuals in an 06 section.

Modular change tasks may be achieved with the gas generator in either the horizontal or the vertical position. Experience in service, however, has demonstrated that maximum benefits can be obtained by using a combination of both. For this reason the tasks have been defined in kits which simplifies the tools and equipment lists and assists the customer in deciding on his particular requirements.

Working areas vary from installation to installation and for this reason procedures differ, but experience has shown that there is generally space on site for all modular change tasks. Failing this it is recommended that the gas generator be removed to either a small local work shop or heavy maintenance base. For total or modular change capability the minimum lifting hook height should be 17 feet with a minimum floor area of approximately 4440 square feet.

All of the tasks can be carried out on a single gantry although two hoists of three tons and two tons capacity will be required to turn the gas generator from the horizontal to the vertical.

In Situ Inspection (Fig 17)

To facilitate internal inspection of the gas generator boroscope ports are provided in all the main casings so that the rotating assemblies can be inspected without removal of the gas generator from the set or indeed without any major strip of the unit.

J.P. Inglis

K.C. Hobbs

Kit No.2 Starter drive

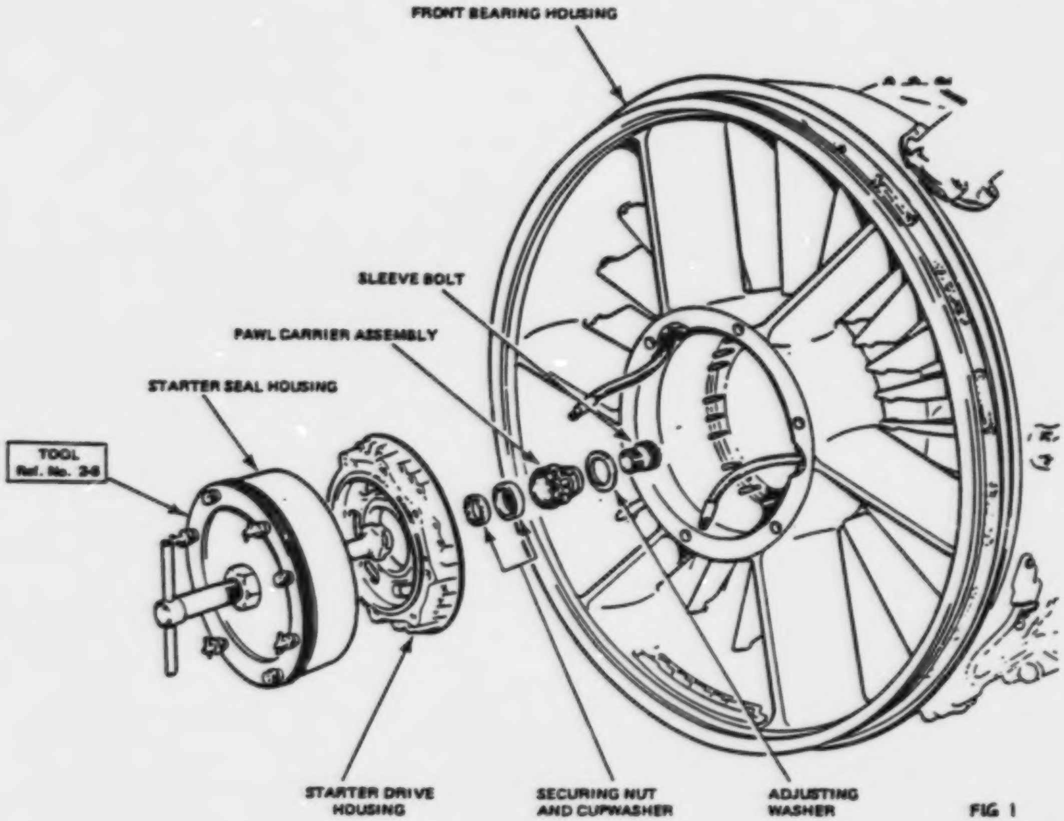

FRONT BEARING HOUSING
SLEEVE BOLT
PAWL CARRIER ASSEMBLY
STARTER SEAL HOUSING
TOOL Ref. No. 2-6
STARTER DRIVE HOUSING
SECURING NUT AND CUPWASHER
ADJUSTING WASHER

FIG 1

FIG. 1

Kit No. 3 Front bearing housing

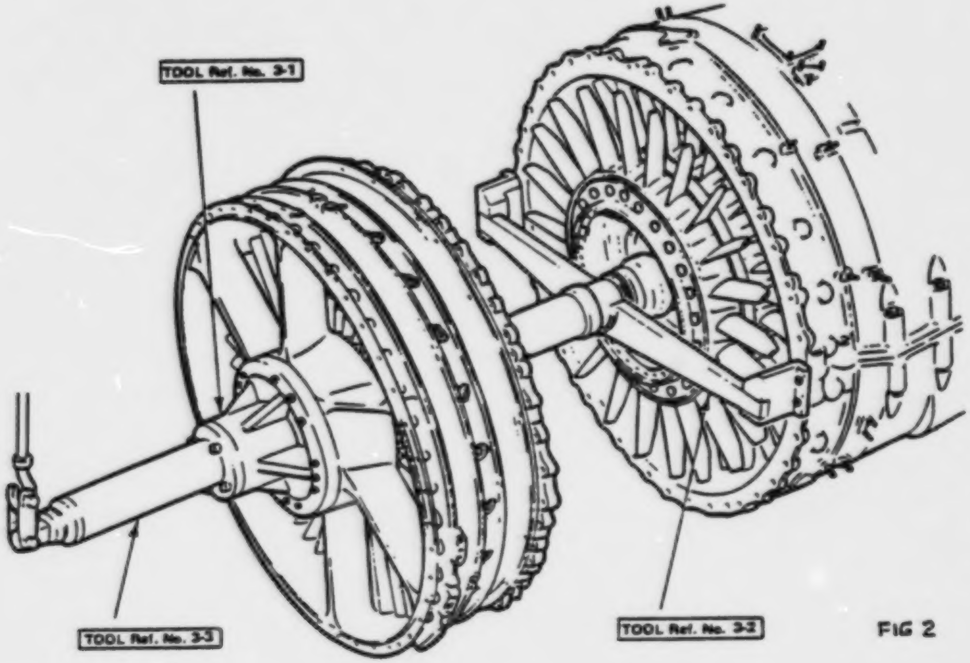
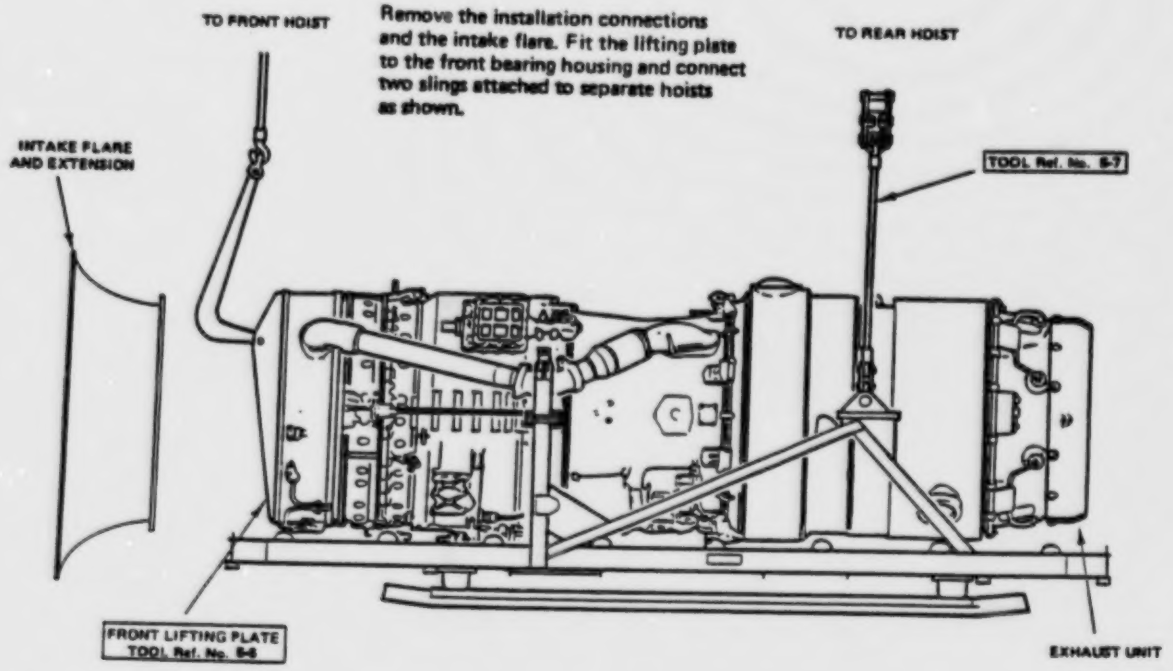

TOOL Ref. No. 3-1
TOOL Ref. No. 3-3
TOOL Ref. No. 3-2

FIG 2

FIG. 2

Kit No. 5 Transferring gas generator from the horizontal to vertical position

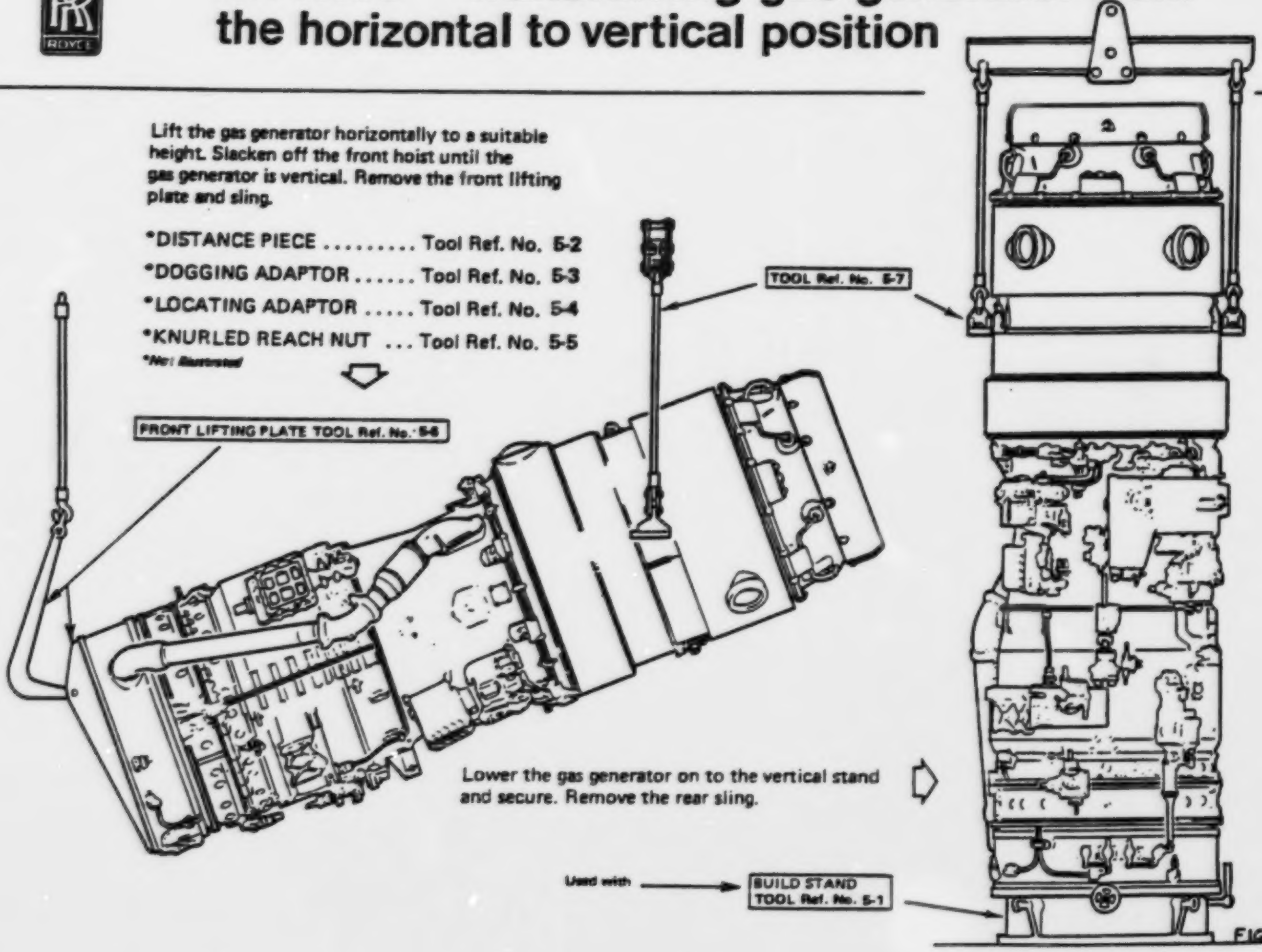

TO FRONT HOIST
TO REAR HOIST
INTAKE FLARE AND EXTENSION
TOOL Ref. No. 6-7
FRONT LIFTING PLATE TOOL Ref. No. 6-6
EXHAUST UNIT

Remove the installation connections and the intake flare. Fit the lifting plate to the front bearing housing and connect two slings attached to separate hoists as shown.

FIG 3

FIG. 3

Kit No. 5 Transferring gas generator from the horizontal to vertical position



FRONT LIFTING PLATE TOOL Ref. No. 5-6
TOOL Ref. No. 5-7
Used with
BUILD STAND
TOOL Ref. No. 5-1

Lift the gas generator horizontally to a suitable height. Slacken off the front hoist until the gas generator is vertical. Remove the front lifting plate and sling.

*DISTANCE PIECE .... Tool Ref. No. 5-2
*DOGGING ADAPTOR .... Tool Ref. No. 5-3
*LOCATING ADAPTOR ...... Tool Ref. No. 5-4
*KNURLED REACH NUT ... Tool Ref. No. 5-5

*Not illustrated

Lower the gas generator on to the vertical stand and secure. Remove the rear sling.

FIG 4

FIG. 4

Kit No.7
Upper and lower half compressor casing

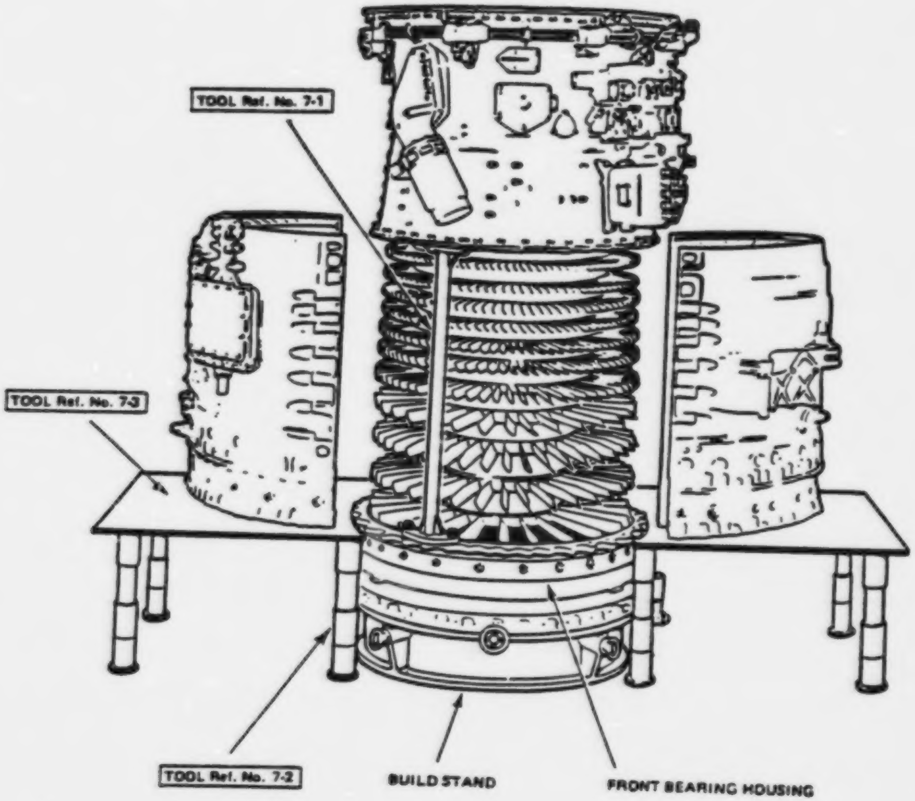

TOOL Ref. No. 7-1
TOOL Ref. No. 7-3
TOOL Ref. No. 7-2
BUILD STAND
FRONT BEARING HOUSING

FIG 5

FIG. 5

Kit No.4
Compressor stage 00 & 0 rotor assemblies

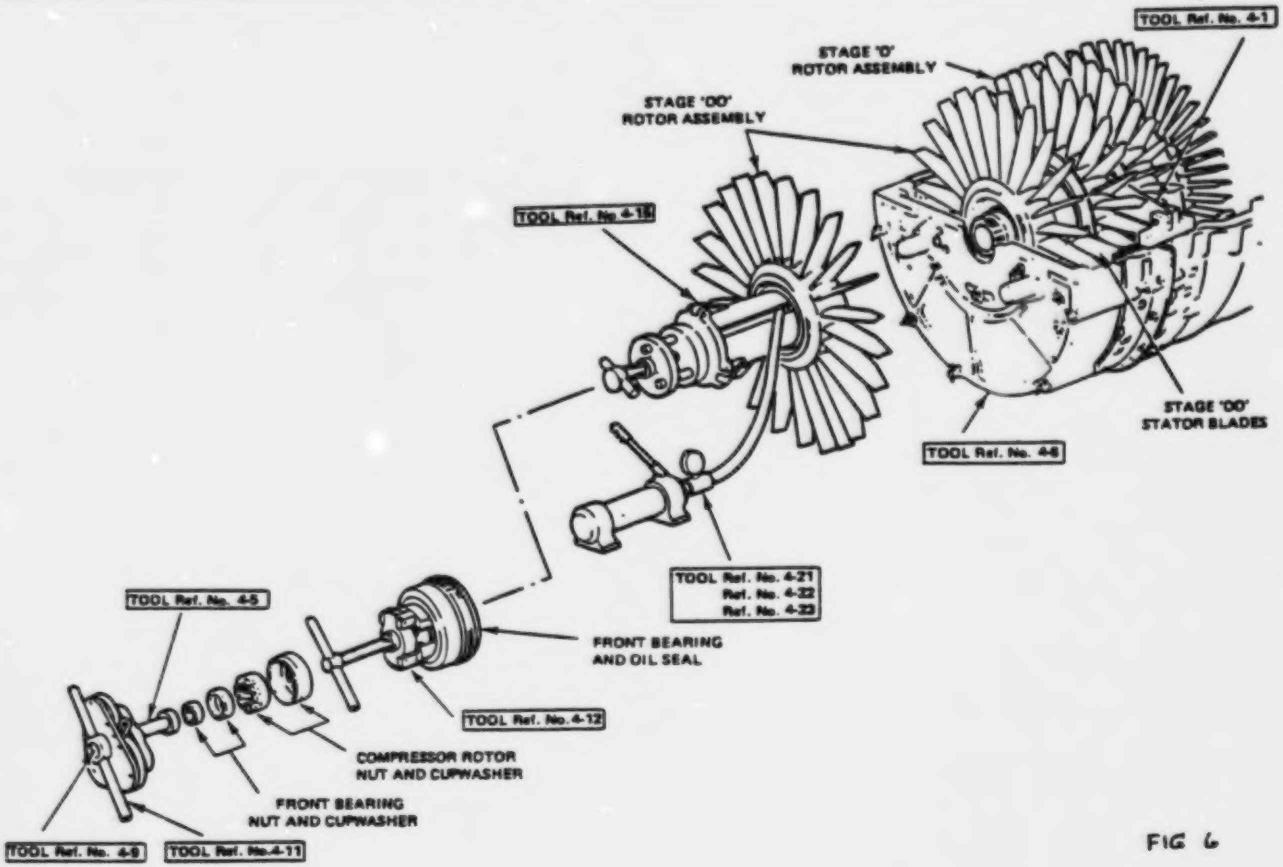

TOOL Ref. No. 4-1
STAGE 'O' ROTOR ASSEMBLY
STAGE 'OO' ROTOR ASSEMBLY
TOOL Ref. No. 4-15
STAGE 'OO' STATOR BLADES
TOOL Ref. No. 4-6
TOOL Ref. No. 4-21
Ref. No. 4-22
Ref. No. 4-23
FRONT BEARING AND OIL SEAL
TOOL Ref. No. 4-5
TOOL Ref. No. 4-12
COMPRESSOR ROTOR NUT AND CUPWASHER
FRONT BEARING NUT AND CUPWASHER
TOOL Ref. No. 4-9
TOOL Ref. No. 4-11

FIG 6

FIG. 6

Kit No.8 Low pressure turbine wheel

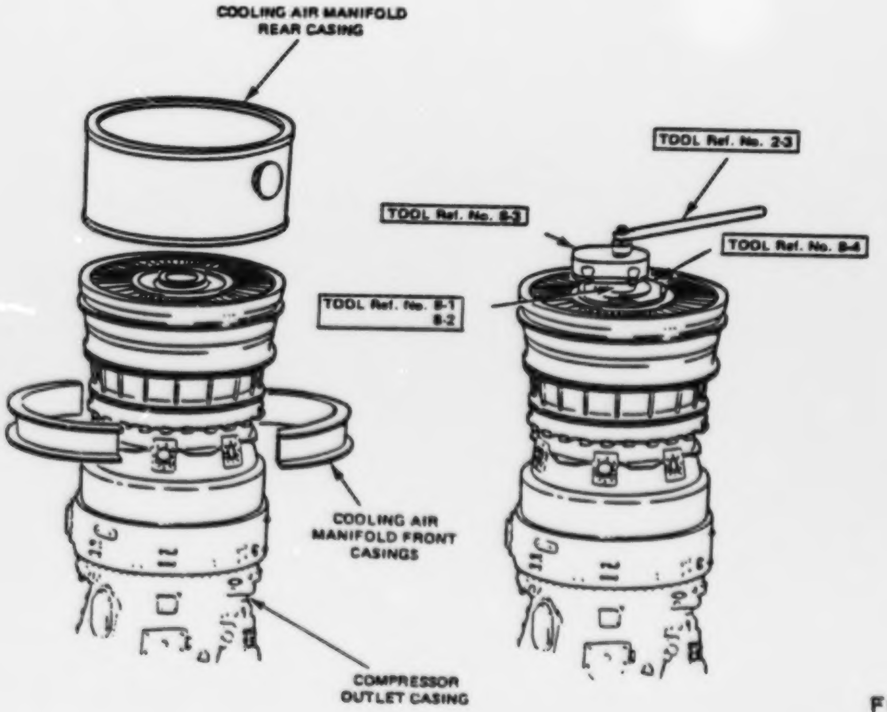
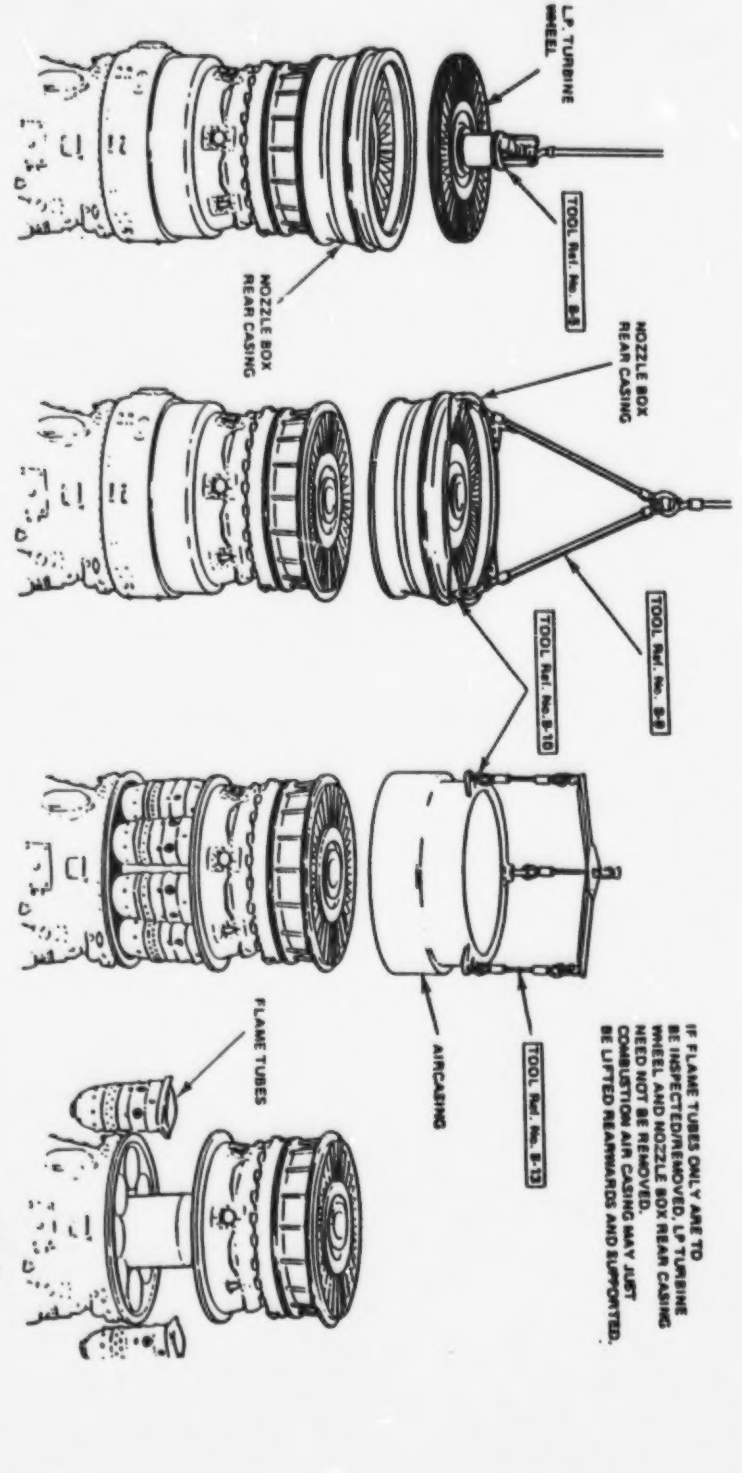

COOLING AIR MANIFOLD REAR CASING
TOOL Ref. No. 2-3
TOOL Ref. No. 8-3
TOOL Ref. No. 8-4
TOOL Ref. No. 8-1
8-2
COOLING AIR MANIFOLD FRONT CASINGS
COMPRESSOR OUTLET CASING

FIG 7

FIG. 7

Kit No.8 Flame tubes


L.P. TURBINE WHEEL
TOOL Ref. No. 8-5
NOZZLE BOX REAR CASING
NOZZLE BOX REAR CASING
TOOL Ref. No. 8-9
TOOL Ref. No. 8-10
TOOL Ref. No. 8-13
AIRCASING
FLAME TUBES

IF FLAME TUBES ONLY ARE TO BE INSPECTED/REMOVED, LP TURBINE WHEEL AND NOZZLE BOX REAR CASING NEED NOT BE REMOVED. COMBUSTION AIR CASING MAY JUST BE LIFTED REARWARDS AND SUPPORTED.

FIG 8

FIG. 8

Kit No.9 Nozzle box and turbine assembly

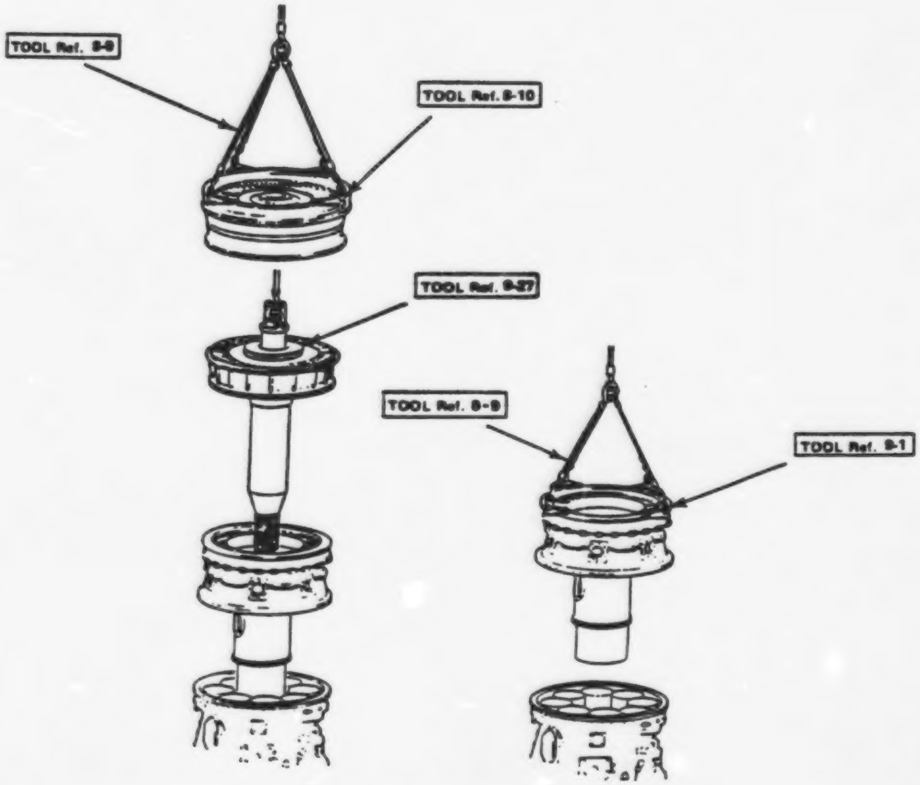

TOOL Ref. 9-9
TOOL Ref. 9-10
TOOL Ref. 9-27
TOOL Ref. 9-9
TOOL Ref. 9-1

FIG 9

FIG. 9

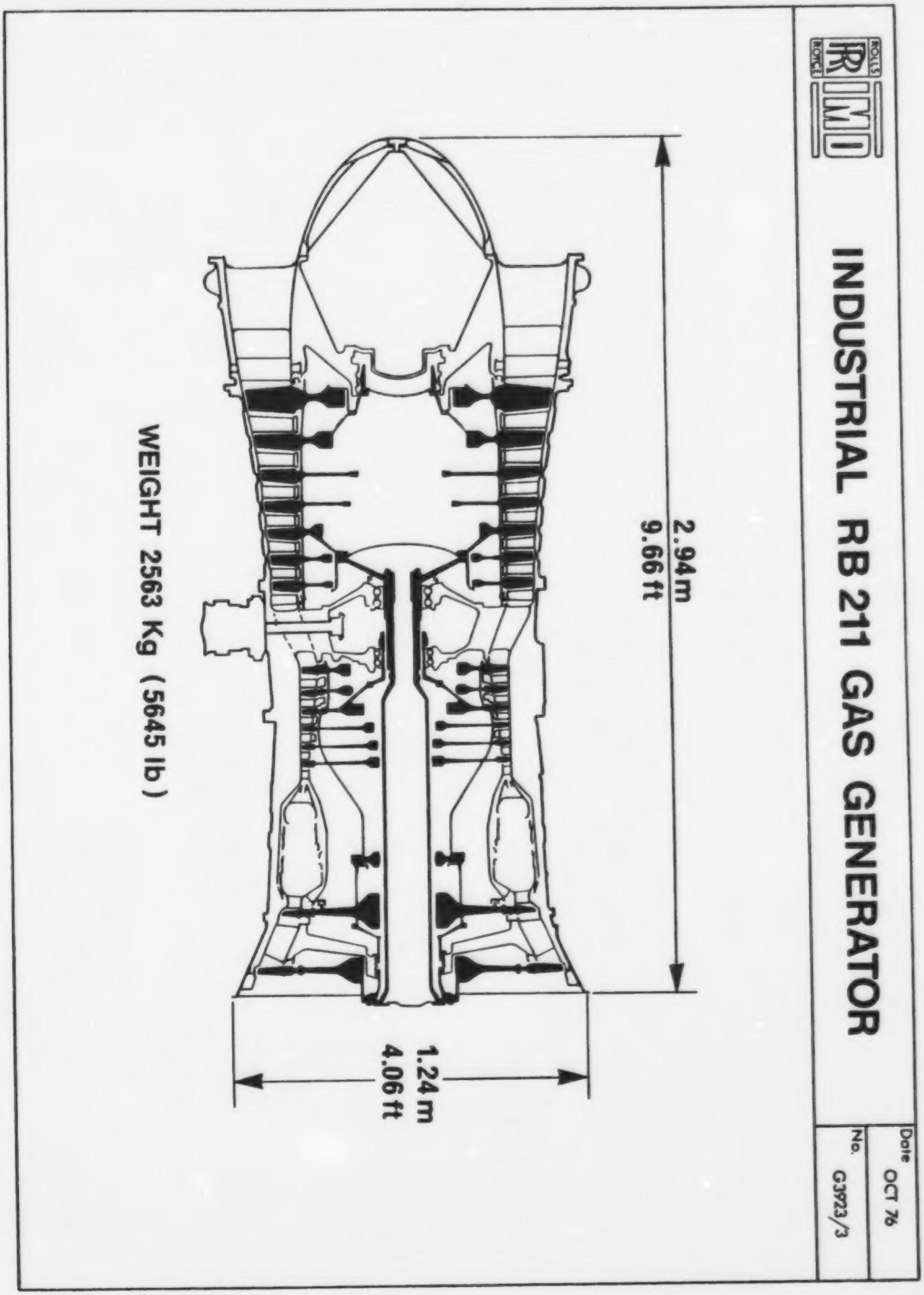

INDUSTRIAL RB 211 GAS GENERATOR
2.94 m
9.66 ft
1.24 m
4.06 ft
WEIGHT 2563 Kg (5645 lb)
Date OCT 76
No. G3923/3

FIG. 10

FIG. 10

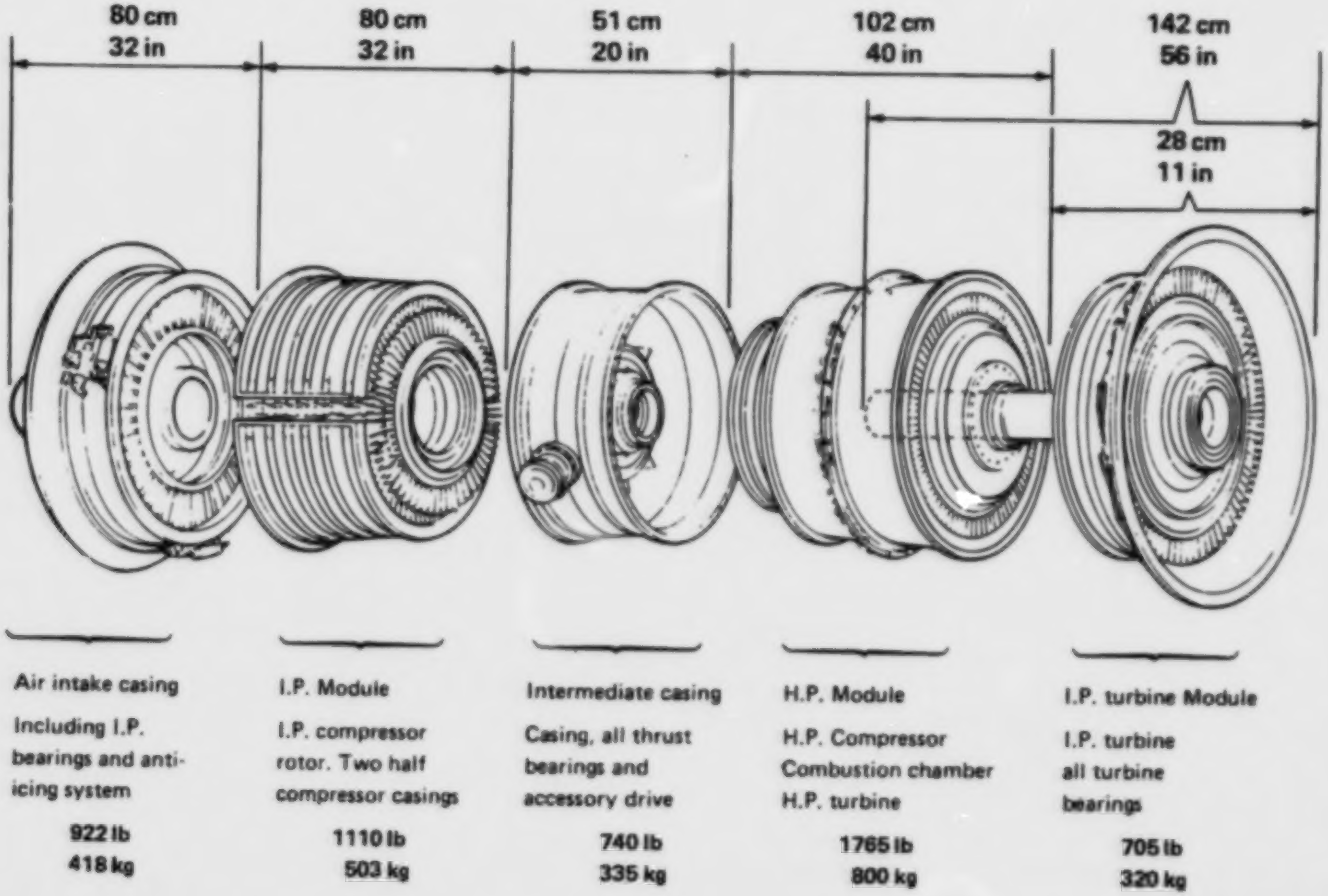

80 cm
32 in
80 cm
32 in
51 cm
20 in
102 cm
40 in
142 cm
56 in
28 cm
11 in
Air intake casing
Including I.P. bearings and anti-icing system
922 lb
418 kg
I.P. Module
I.P. compressor rotor. Two half compressor casings
1110 lb
503 kg
Intermediate casing
Casing, all thrust bearings and accessory drive
740 lb
335 kg
H.P. Module
H.P. Compressor
Combustion chamber
H.P. turbine
1765 lb
800 kg
I.P. turbine Module
I.P. turbine
all turbine bearings
705 lb
320 kg

FIG. 11

F16 11

RB211 CHANGE MODULE 01

Date

No. G5297/1

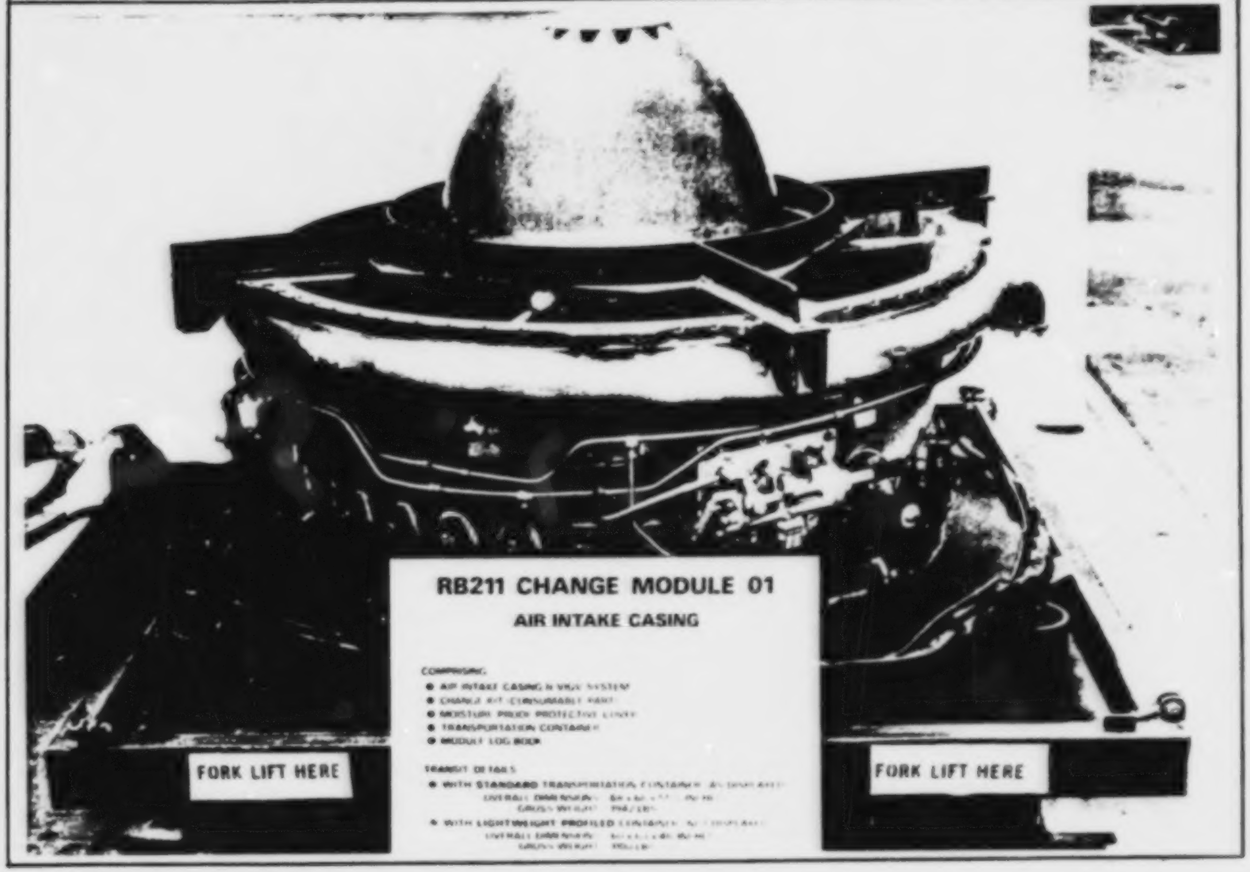
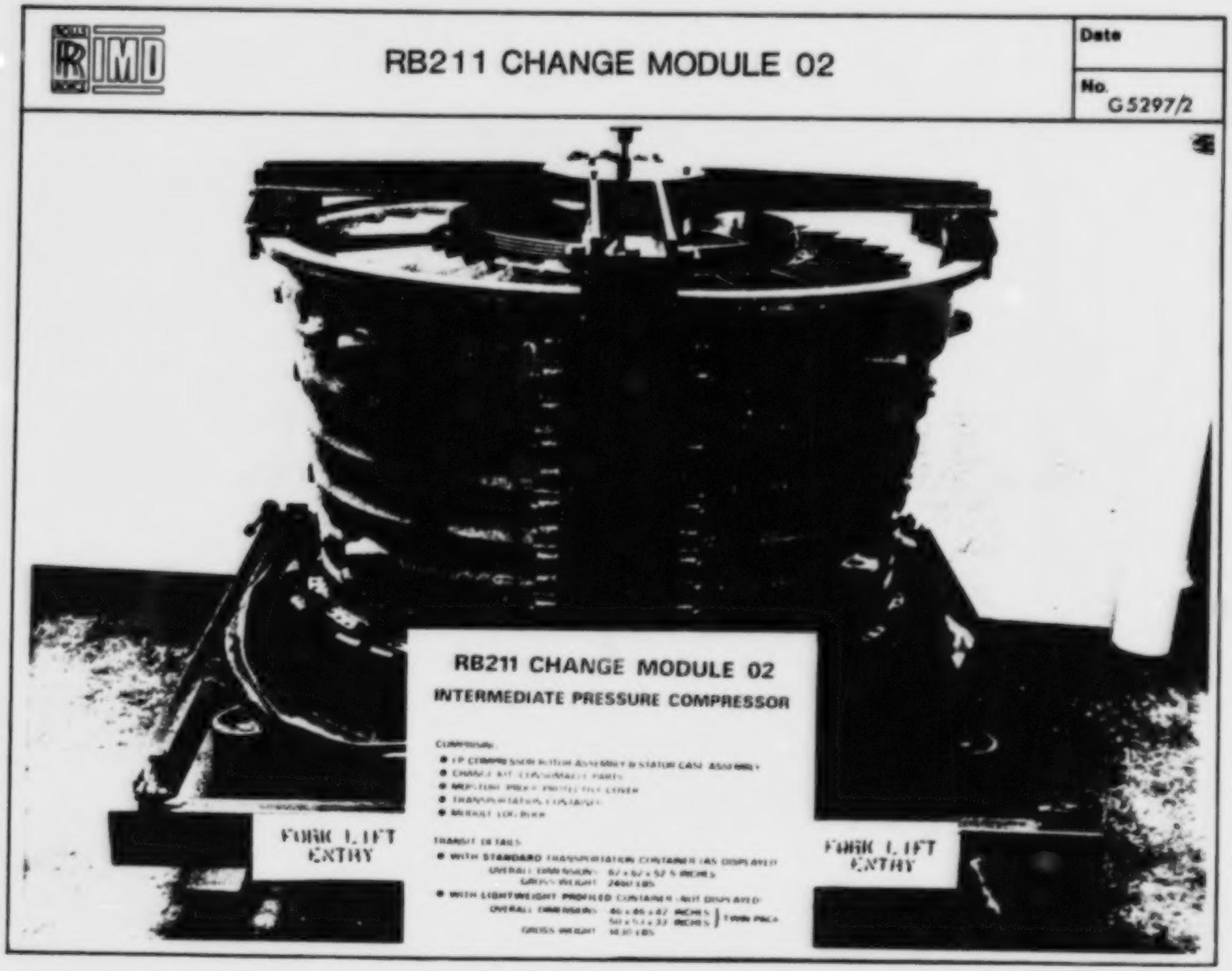

RB211 CHANGE MODULE 01
AIR INTAKE CASING
FORK LIFT HERE
FORK LIFT HERE

FIG. 12



RB211 CHANGE MODULE 02

Date

No. G5297/2

RB211 CHANGE MODULE 02
INTERMEDIATE PRESSURE COMPRESSOR
FORK LIFT ENTRY
FORK LIFT ENTRY

FIG. 13

RB211 CHANGE MODULE 03

Date

No. G5297/3

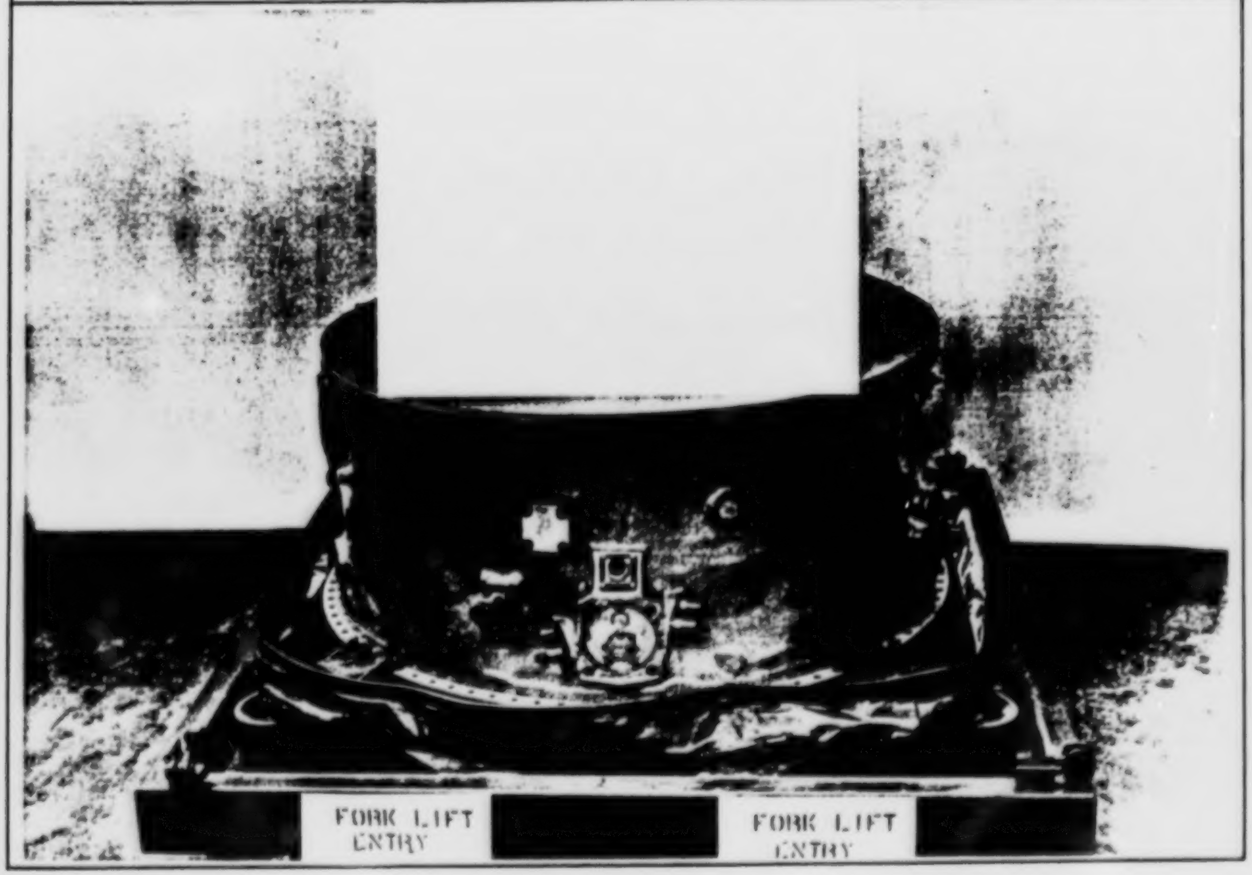
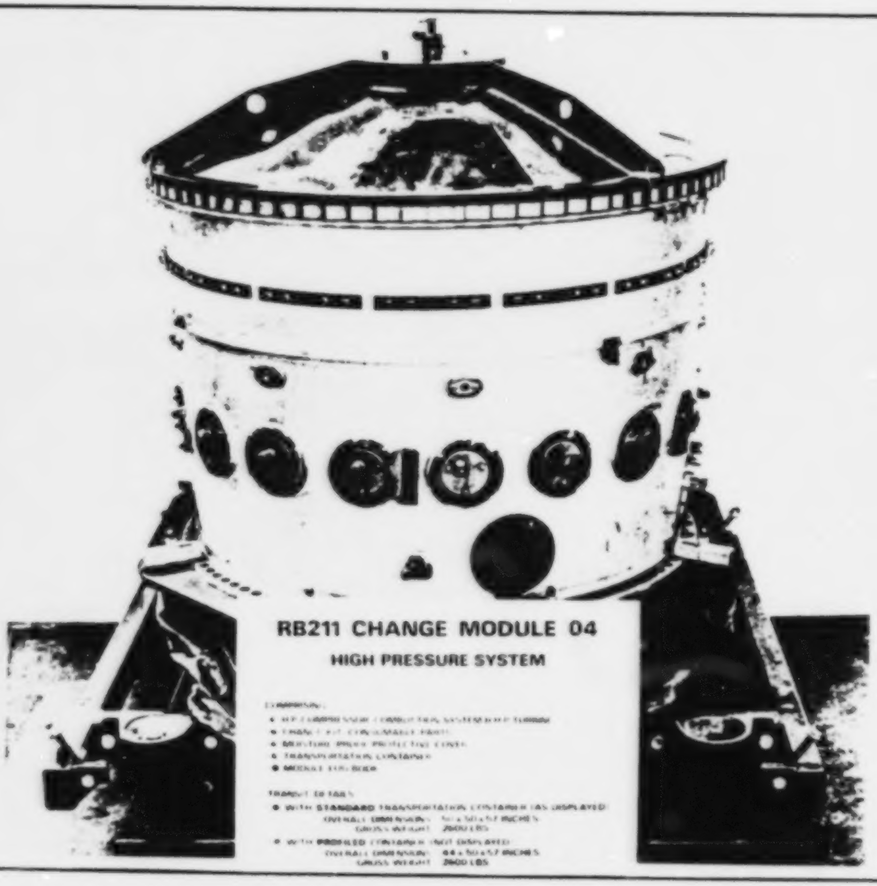

FORK LIFT
ENTRY
FORK LIFT
ENTRY

FIG. 14

RB211 CHANGE MODULE 04

Date

No. G 5297/4



RB211 CHANGE MODULE 04
HIGH PRESSURE SYSTEM

FIG. 15

RB211 CHANGE MODULE 05

Date

No. G 5297/5

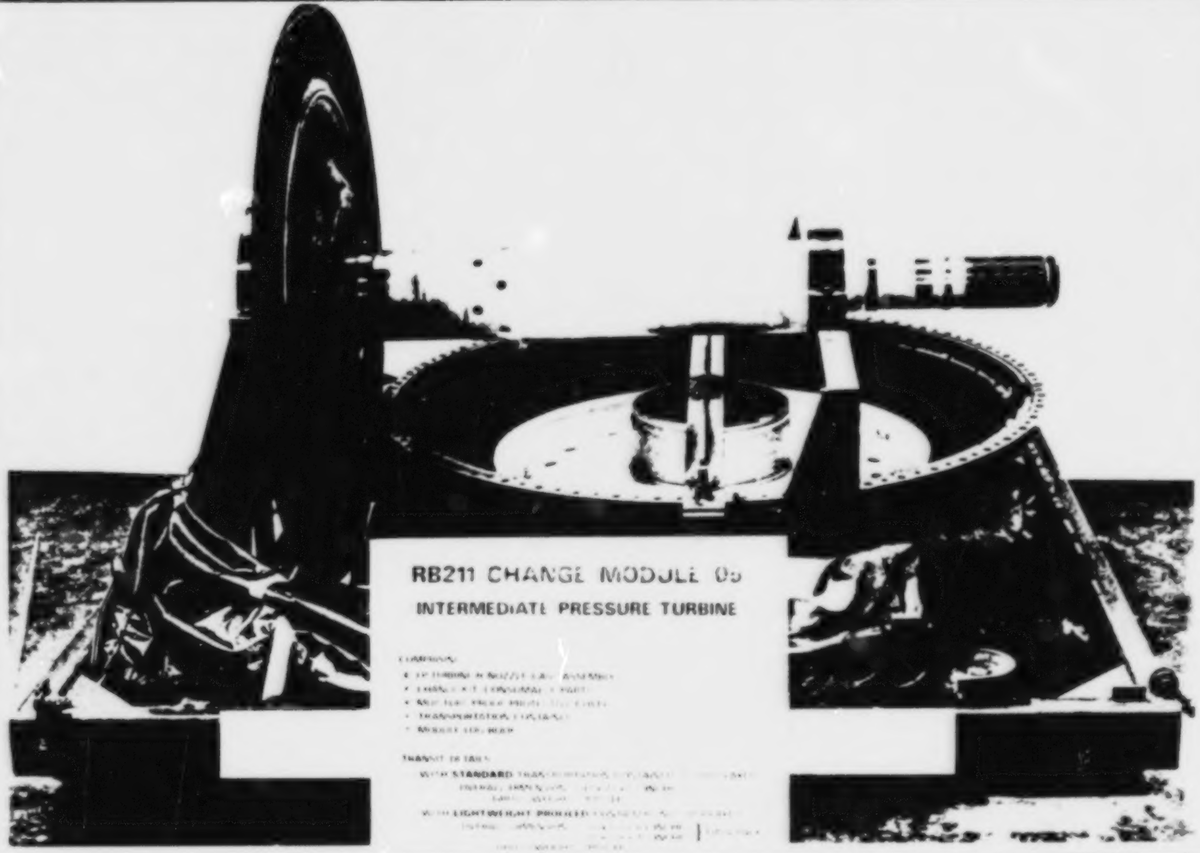
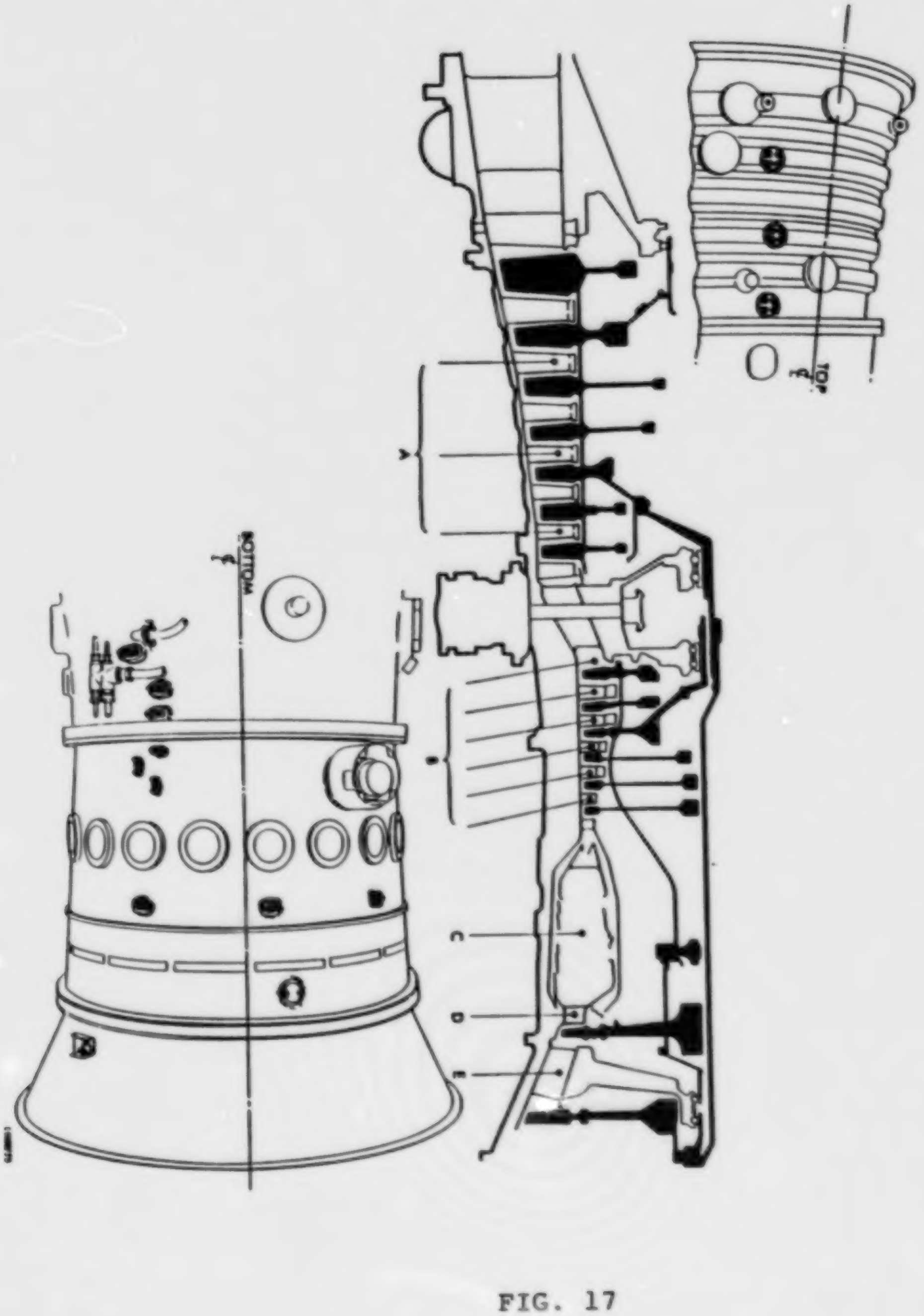

RB211 CHANGE MODULE 05
INTERMEDIATE PRESSURE TURBINE
STANDARD
LIGHTWEIGHT

FIG. 16


A
B
C
D
E
TOP ℄
BOTTOM ℄

BOROSCOPE ACCESS POINTS

FIG. 17

LAGOVEN S.A.
IN THE WESTERN DIVISION OF VENEZUELA
GAS TURBINE MAINTENANCE EXPERIENCE

LAGOVEN S.A.
IN THE WESTERN DIVISION OF VENEZUELA
GAS TURBINE MAINTENANCE EXPERIENCE

INTRODUCTION

LAGOVEN S.A. is one of the affiliates of Petroleos de Venezuela, the state-owned company responsible for all aspects of the Oil business in Venezuela. Its Western Division handles a major portion of the production operations and its potential is 917.000 BPD, 37% of the National production capacity.

In this division, the gas turbine is the principal prime mover, taking advantage of large resources of natural gas to provide power for the pressurization and transport of fluids (natural gas, crude, water, etc.) and for the generation of electrical power.

The present paper describes the division's experience in the operation and maintenance of gas turbines.

OPERATIONAL ENVIRONMENT

Most of the facilities are located offshore in Lake Maracaibo, on platforms 14 feet or more above lake level. An appreciable salt content in the air due to wind pick-up from the brackish lake water is consequently an important environmental feature.

Ambient temperature remains almost constant the whole year round, ranging between 80 and 95°F. During the year, there are two dry seasons (January - March and July - September) with considerable wind blown dust from the adjacent shore, and two rainy seasons (April - June and October - December), which influence particularly the performance deterioration of axial compressors.

The natural gas fuel is clean, with low levels of contaminant - 2% CO_2, but H_2S content is increasing in some areas (up to 50 ppm).

APPLICATIONS

The early offshore plants, commencing in 1954 with Tía Juana 1 Gas Conservation Plant (a milestone plant in its time, referred to in Marks Mechanical Engineers Handbook), utilized heavy duty industrial gas turbines.

Since aeroderivative turbines became available in the 60's however, the large majority of offshore plants, as well as remote land stations, have utilized this type of turbine, taking advantage of the reduced plattform structural requirements and low on-site maintenance. Such plants include Water Injection Units and Miniplants for gas compression on the lake, as well as booster pumping stations along the 230 Km crude pipeline between the Lake's eastern shore and the Amuay Refinery, at a deep water terminal on the Caribbean.

The division's power plant at Punta Gorda utilizes industrial turbines, including four 20MW turbo-generator packages installed since 1984 to increase plant capacity.

Table 1 shows the different turbine models operating in the division together with application details.

TABLE 1

APPLICATIONS IN THE WESTERN DIVISION

MODEL	TYPE		YEAR FIRST UNIT STARTED	MAXIMUM HOURS ACCUM. ON ONE UNIT	SERVICE
	HEAVY DUTY	AERO GGs/PTs			
GE MS-3002	10		'54	240k	Gas Compression
West W101	42		'57	210k	Gas Compression
Orenda OT-3		16/10	'72	30k TBO	Water Injection and Oil Pumping
RR Avon MK 1534/ Cooper RT-48		16/10	'73	40k TBO	Gas Compression
Ruston TB-1750	5		'70	130k	Water Injection
Solar Saturn	4		'73	100k	Gas Compression
Solar Centaur	5		'73	100k	Propane Comp. for Refrigeration
RR RB-211/ Dresser DJ-290R		5/4	'85	20k	Gas Compression
GE LM-2500/ Dresser DJ-270R		5/5	Commissioning	0	Gas Compression
GE MS-7001	2		'59	200k	Power Generation
BBC Type 10-BV/65000	1		'67	150k	Power Generation
GE MS-5001 P	4		'84	26k	Power Generation
TOTALS	73	42/29			
TOTAL GAS TURBINES:	102				

LAGOVEN has accumulated 33 years of continuous operational experience on gas turbines. To allow 24 hours a day service, mostly in unattended operation, a modern and integral maintenance program is required, which involves failure prevention as well as the logistics to put units back in service as soon as possible after failures occur. The objective of this effort is to maintain high operational indices. Table 2 shows availability and reliability factors, together with service and load factors for reference. It should be noted that these figures refer to the whole installation, and not just the gas turbine.

TABLE 2

	RELIABILITY		AVAILABILITY		SERVICE F.		LOAD FACTOR	
	85	86	85	86	85	86	85	86
WESTINGHOUSE W101	99.2	99.2	96.5	96.5	81.3	90.7	100.0	91.2
GE MS 3002	97.9	98.9	97.2	98.1	62.0	98.1	93.0	91.4
AVON/COOPER	93.0	92.0	82.4	87.0	40.0	76.0	100.0	100.0
ORENDA	92.0	86.0	91.0	85.0	76.5	75.8	80.0	75.0
RUSTON	95.0	93.0	87.0	87.9	82.0	85.0	92.0	92.0
GE MS5001	94.0	97.0	93.0	95.0	83.0	80.0	90.0	90.0

MAINTENANCE PHILOSOPHY

LAGOVEN'S approach to maintenance involves, as for many operators, preventive measures to avoid the development of problems or to detect them in time to avoid major failures. Also, considerable flexibility has been developed to minimize down time when failures or other factors prevent unit operation. By design, the gas compression, water flood, power and oil pumping systems have generally a degree of redundancy, compensating to some extent the consequences of unavailability of any individual unit. Due to the relative isolation of Venezuela from reliable sources of maintenance services, it was considered essential to build up in-house resources to accomplish all except the most specialized maintenance activities, and services were developed that many users would contract out, as will be shown below.

LAGOVEN'S TURBOMACHINERY SHOP

The turbomachinery shop is located in La Salina, one section of the company's large Central Workshop with a total area of 10^6 sq. ft.

This shop handles a great variety of work, such as rotor repair and balancing, repair of hot gas path components, overhaul of centrifugal compressors, steam turbines and high pressure multistage pumps, as well as the repair of aeroderivative gas turbines.

Some details of the shop available machinery capacity, are presented in Table 3.

TABLE 3

TURBOMACHINERY SHOP CAPABILITIES

SECTION	CAPACITY
Machine Tool	14 Horizontal Lathes
	1 Vertical Lathe
	2 Milling Machines
	3 Boring Machines

	3 Planers
	1 Broaching Machine
	2 Bench Saws
	1 Swing Saw
	3 Grinders
	1 Tool Grinder
Welding	Tungsten Welding
	Inert Gas Welding
	Arc Welding
Cranes	2 Bridge Cranes (100 Ton)
	10 Gantry Cranes (20-50 Ton)
	78 Jib Cranes (1-2 Ton)
	2 Gantry Cranes (5 Ton)
Cleaning	Chemical Cleaning
	Steam Cleaning
	Sandblasting
	Metalblasting
Inspection	X Rays
	Magnaflux
	Zyglo Testing
	Penetrant Red Dye
	Profile Comparator
Balancing	2 Horizontal Balancing Machines (Max. 28 Ton)
	1 Static Balancing Machine (Momentum)
Mounting and Heat Treatment	Hydraulic Press (150 Ton)
	Freezer (-135°C)
	Electric Furnace

FIELD MAINTENANCE ORGANIZATION

Experienced maintenance teams, with personnel involved in mechanical maintenance, and in other crafts, such as instrumentation, construction and electrical maintenance, have been built up to carry out the field repair of rotating equipment. One example of the ability developed, is the normal completion of a Westinghouse W101 overhaul in eight days, including rotor exchange and field balance.

The maintenance organization includes a Preventive Maintenance Group, whose role is to formulate and control preventive maintenance programs as will be described later.

OPERATIONAL ORGANIZATIONS

The operational organizations have an important role in the control of operating and maintenance costs, and have the final decision on when to carry out programmed maintenance activities, according to production requirements.

For instance, the operating organization for the Gas Conservation Plants, which uses 86% of the division's gas turbines, collects all the information concerning production targets, maintenance requirements and preventive inspections and generates a master yearly maintenance plan. The plan is flexible and serves as a guide to update on a monthly basis a program for the following three months, which takes into account fluctuations of production requirements due to market changes and unplanned maintenance needs. The organization also emits a monthly report covering important operations and maintenance information, including operating hours of gas turbines and some of their critical components, and the implementation status of all applicable recommendations.

ENGINEERING SUPPORT

Prior to the industry's nationalization in 1976, engineering support was mostly provided by specialists from overseas. In 1977, a technical advisory group was formed within the General Engineering organization, to compensate the lack of specialized consultants in the Venezuelan market, and to anticipate the termination of consultancy agreements with the previous operating companies.

This advisory group covered initially Rotating Equipment Engineering, Materials Engineering, and Instrumentation and Control Engineering. Subsequently existing sections for Corrosion Control and Equipment Inspection were merged into the group.

The role of the group is to maximize the availability and efficiency of the facilities, eliminating critical and recurrent problems and introducing new technologies. Its impact has been very significant since many problems suffered, despite efforts by foreign consultants, during 20 years or more, were finally corrected by this group, with collaboration from the equipment manufacturers.

Through the group LAGOVEN maintains an exchange of technical information with many rotating equipment users, manufacturers and consultants.

TYPES OF MAINTENANCE

LAGOVEN carries out three types of maintenance:

- Preventive maintenance (statistically based)
- Condition-based or predictive maintenance
- Corrective maintenance

In unattended locations, using aeroderivative turbines, a preventive maintenance program is applied to the whole facility. In a typical plant such as a Waterflood unit or a Mini compression plant, the preventive maintenance program includes two levels of maintenance as shown in Table 4.

TABLE 4

	FREQUENCY (MONTHS)	
LEVEL	MINI PLANT	WATER FLOOD UNIT
Minor	2	3
Major	24	18

Gas turbine and other rotating equipment auxiliary systems, are included within the maintenance program. At the minor level, work includes recalibration of instrumentation, changeout of filter elements, regreasing, etc. At the major level, more complicated work is performed such as structural and piping repairs, valve and actuator servicing, preventive changeout of coolers, etc. The gas turbines themselves receive preventive inspections on a fired hour basis, but as far as possible these are adjusted to coincide with preventive maintenance shutdowns. Table 5 shows inspection intervals applied to aeroderivative engines, based on manufacturer's recommendations corrected in the light of local experience.

TABLE 5

	FREQUENCY-FIRED HOURS		
TYPE	0-0 STG. INSP.	GENERAL INSPECTION (IN SHOP)	OVERHAUL
ROLLS-ROYCE AVON	10.000	20.000	40.000
ORENDA OT-3	-	6.000	25.000

In attended facilities, such as the Conservation Plants, the turbine determines the maintenance action. Table 6 shows the inspection and overhaul intervals.

TABLE 6

	INSPECTIONS		
	COMBUSTION	HOT GAS PATH	OVERHAUL
GE MS 3002/MS 7001	8000	-	50000
WESTINGHOUSE W101	8000	-	40000
GE MS-5001	8000	24000	48000*

*The 1st overhaul has not yet been performed.

During the 8000 hours inspection, all instrumentation is recalibrated, combustors are usually changed and incipient problems in auxiliary equipment are corrected.

Among the heavy duty industrial gas turbines population, two overhaul practices are followed:

The MS-3002 based plant, features a complete spare turbine, which is rotated to permit removal of each turbine in turn for overhaul. Rotor and hot gas path components are repaired in the La Salina Turbomachinery Shop and other repairs are effected on site.

The fleet of 42 Westinghouse turbines features six complete spare rotors, which are replaced in the field. The large number of turbines makes this an important on-going activity, requiring annually approximately 12 overhauls and 30 combustion inspections, and usually some additional effort due to unpredicted failures.

As part of the effort to reduce maintenance costs, considerable emphasis is placed on extending the useful life of the turbine components. For instance, the Westinghouse fleet, which achieves high reliability factors due to the progressive elimination or atenuation of recurrent problems and the high degree of experience and practice accumulated over the years in the maintenance activities, are running with blades that have exceeded the maximum running hours recommended by the manufacturer, based on an evaluation which was made on a population of used blades to determine new life limits. Based on this evaluation, Lagoven, at its own risk, has decided to extend the operational limits of the Westinghouse blading to those indicated in Table 7.

TABLE 7

RETIREMENT LIMITS FOR WESTINGHOUSE W101 BLADES

		MAXIMUM PERMITTED OPERATING HOURS	
STAGE	ALLOY	ORIGINAL	CURRENT
1	INCO 700	100.000	150.000
2	INCO 700	100.000	150.000
3	INCO X 750	100.000	120.000
4	INCO X 750	100.000	130.000
5	INCO X 750	100.000	170.000

More recently, some HIP tests were made by an external consultant on 3rd and 4th stage blades with successful results, indicating the possibility of further extension of the blade life for these stages. Estimated savings of the order of 600 M$/year are obtained from this blade life extension activity.

PREDICTIVE MAINTENANCE

The Rotating Equipment Section of the technical advisory group, have put effort into mastering the use of vibration analysis techniques, as a diagnostic tool for predicting machine condition. Routine measurements at appropriate intervals have been established in some areas, as a basis for trend monitoring and the prediction of proper maintenance timing.

Avon gas generators have become a critical area, due to the strategic importance of the Mini compression plants. In this area, the predictive maintenance activities have been expanded and have made an important contribution to reducing maintenance costs, together with the improvement of the quality of local repairs. Boroscope inspections are performed to avoid unnecessary removals. Monitoring of the intake pressure drop of the axial compressor is being applied to determine when to effect crank-soak washing of the compressor so as to maintain performance levels. A methodology for performance evaluation, adapted to the specific equipment, has been developed, and is currently in use. Routine vibration analysis is also performed on these engines.

CORRECTIVE MAINTENANCE

Despite efforts to maintain reliability, failures still do occur. All turbine types have experienced recurrent problems and many of them have been partially or totally solved. Some of the outstanding situations on two typical machines of Lagoven's fleet, the Westinghouse W101 and Rolls Royce Avon gas generator are noted in the following.

- Turbine Disk Cracks on Westinghouse W101 Turbines

This problem, of great concern in the past, has now been reduced to a minimum, due to various modifications made by the manufacturer. When present, failures are always detected during the 40000 hour inspections as incipient cracks. Although they have had no effect on the availability of the turbine, the influence on costs has been considerable, since a disk costs about $120,000.

Historical experience of the problems is summarised briefly in the following:

4th Stage: 1965. High Cycle Fatigue. Alleviated by the introduction of rebated blades.

2nd Stage 1971. Creep, following uprate from model W81 to W101 causing disk failure after 2000 hours. Alleviated by the addition of cooling holes.

5th Stage 1977. Creep and High Cycle Fatigue. Alleviated by optimization of disk alloy and the installation of new 5th stage blades for higher speed machines.

Cracks are still occasionally found at overhaul in the disks of any of the five stages, but have never been of a dangerous length. We still have some operation with high disk metal temperature and with flow pulsations due to leaks at stator horizontal joints in the turbine, which undoubtedly has some influence on these results.

Similar disk-cracking has been found in the first stage discs of Cooper RT48 power turbines, and some corrective actions, including a new blade holding design, are currently being implemented.

- Avon Gas Generator Damage Due to Inadequate Air Filtration

The lack of an adequate air filtration system in the original plant design has been an important factor in accelerated deterioration of the axial compressor, resulting in considerable performance reduction from deposits, blade encastoring and subsequent failures, and major corrosion damage.

A great improvement has been obtained by the implementation of a state-of-the-art filtration system. The economic benefit of the new system has been estimated and has been used to justify an accelerated program of filter house replacement for all the Mini Plants.

Another recurrent problem on these GG's is cracking of axial compressor casings. External corrosion on joints of the forward support bracket, prevent the natural expansion of the GG, inducing undue stresses. An additional factor which has been recently discovered and may be influential in the casing cracking problems is an effect of non uniform cooling with the present cabin ventilation arrangements. A high vibration peak at half times running frequency comes and goes when the left ventilation fan is started and stopped. This phenomenon has been associated with thermal distortion of the casing and a resulting tip rub. The quantitative effect on the casing cracking problem remains to be determined however.

CONCLUSIONS

LAGOVEN has been operating and maintaining gas turbines for 33 years. During this time, in-house capacity has been progressively developed to overcome the relative isolation from reliable sources of maintenance assistance.

A comprehensive maintenance program applied to these engines, including close follow-up of the operational indices to maintain them under control, has been a key factor in the satisfactory results achieved.

DEVELOPMENT AND APPLICATION OF DIFFUSION BRAZE REPAIR TECHNIQUES

S.G. Stoykewich - Manager, Repair and Overhaul
R.J. Armstrong - Metallurgical Engineer

VAC-AERO INTERNATIONAL INC.,
1371 Speers Road,
Oakville, Ontario.
L6L 2X5

SUMMARY:

Repairing of turbine engine nozzle guide vanes with diffusion braze alloys is a viable alternative to conventional repair techniques. The braze joint strength obtained with nickel base diffusion braze alloys on cobalt base (X-40) superalloy specimens was exceptional. Additionally, the joints display a greater degree of ductility relative to conventional nickel base non-diffusion braze alloy joints. When blended with base material alloy powder, the diffusion braze alloys are capable of brazing substantially large voids. The ability to apply a diffused aluminum-chromium coating on diffusion braze alloy repaired components was demonstrated. These capabilities will allow a repair agent to recover a greater percentage of turbine components.

Similar results are predicted during repair of gamma prime strengthened nickel base superalloys. Unique to these alloys is the proposed method of cleaning; activated fluoride oxide reduction (AFOR).

1. INTRODUCTION:

In today's market, the emphasis is for development of turbomachinery that operates at higher efficiencies (lower specific fuel consumption). This equates simply to engines performing at increasingly higher temperatures. As operating temperatures increase, material requirements become increasingly critical. Conventional carbide strengthened cobalt superalloys can no longer withstand the operating environment of a high pressure turbine nozzle guide vane. Gamma prime γ' strengthened nickel base superalloys (RENE 80, MAR-M246, IN-738) are gradually replacing the cobalt alloys as turbine inlet temperatures increase. Additionally, the new series of turbine nozzle guide vanes are units cast with hollow cores. Subsequent electro discharge machining and/or laser drilling operations are performed to allow core cooling air to escape and form protective boundary layers on the airfoil surfaces. While these changes have provided the turbine manufacturers with improved operating efficiencies, successful repair of subsequent service damage has been limited. Manual weld repairing of thin wall nozzle guide vanes often produces unacceptable levels of distortion and the majority of the high strength base alloys are considered unweldable. Conventional nickel-chromium-boron-silicon (Ni-Cr-B-Si) braze alloys do not provide the necessary joint ductility for repairing the highly thermal stressed nozzle guide vanes. Due to these limitations, the turbine component repair industry is experiencing a resurgence of interest in the development of improved brazing alloys and techniques.

1. INTRODUCTION: (Cont'd)

The object of this paper is to introduce diffusion brazing as an alternate process for the repair of cobalt base turbine nozzle guide vanes (N.G. Vanes). The paper also sets the scene against which to consider a series of papers that investigates repair of gamma prime strengthened nickel base superalloys. Activated fluoride oxide reduction (AFOR) is introduced as a proposed pre-braze cleaning technique.

2. CONVENTIONAL REPAIR TECHNIQUES:

Repair of service damaged turbine nozzle guide vanes is accomplished by manual tungsten inert gas (T.I.G.) welding or furnace brazing. A combination of the two (2) repair processes may also be performed on a single component. In either case, the repair performance is dependent upon the filler material (weld wire or braze alloy) utilized. In manual welding, the filler material composition often matches that of the component base material. Brazing is typically performed with Ni-Cr-B-Si braze alloys. Deciding on which repair technique to utilize is quite fundamental. The three (3) major factors to consider are the component design, accessibility of repair area and component base material. Component design may be divided into two (2) distinct categories; first-generation N.G.V.'s and second-generation N.G.V.'s. The first-generation N.G.V. is a simple, robust casting with relatively heavy wall sections. Its cooling system, if present, is simple and will continue to function if there is some intrusion of weld metal during repair. The moderate turbine inlet temperature produces a thermally induced stress which, although sufficient to initiate cracking, is low enough for the crack not to propagate rapidly. Therefore, weld repairing with medium strength filler materials is sufficient to last from one overhaul to the next. An example of a first-generation N.G.V. is provided in Figure 1.

A typical second-generation N.G.V. is shown in Figure 2. In comparison with the first-generation component, it is of much lighter section, with a more sophisticated cooling system and operates at higher turbine inlet temperatures. These differences render the second-generation N.G.V. unweldable. The light wall sections distort under heavy welding. The cooling system, being more complex, cannot function as required if penetrated by weld metal during repair. Additionally the higher operating temperatures cause heat affected zone cracks to propagate at alarming rates.

Accessibility of the repair area is an obvious problem for manual weld repairing. Braze repairing is often the only option available, especially when attempts are made to repair wide chord or multi-vane N.G. Vanes.

Presented in Table I are chemical compositions of several cast alloys that are common to turbomachinery N.G. Vanes. Whether the alloy is cobalt or nickel base is of importance to the repair vendor.

The cobalt base superalloys obtain their strength from solid solution hardening of elements such as tungsten and chromium and by precipitation of carbides, primarily chromium carbides. As a result, the alloys are readily weldable and furnace brazed. Similar to these alloys with regard to strength and repairability are the gamma γ strengthened nickel base alloys such as Hastelloy X, Inconel 625 and Inconel 718.

2. CONVENTIONAL REPAIR TECHNIQUES: (Cont'd)

Unfortunately, these alloys are not capable of withstanding the adverse operating conditions of modern turbomachinery N.G. Vanes. Their primary use is in construction of combustion chambers (Hastelloy X) and turbine frames (Inconel 718).

Today's nickel base turbomachinery N.G. Vanes are cast from γ' ($Ni_3(Al, Ti)$ strengthened alloys. Repair of these components is limited to manual welding with lower strength γ filler materials (Inconel 625) or by furnace brazing with conventional alloys. Successful welds in γ' alloys with γ' filler materials are rare. During welding, the precipitation of γ' occurs rapidly and in such an amount that the resulting shrinkage stresses are sufficient to initiate cracks in the weld heat affected zone. Therefore, due to their unweldability and often limited access to repair areas, furnace brazing is often the only repair option available to the vendor.

3. CONVENTIONAL VS. DIFFUSION BRAZE ALLOYS:

During the early sixties, vacuum furnace brazing was proposed as an alternate method for repair of service damaged turbine N.G. Vanes. The braze alloys evolved into what are known as conventional Ni-Cr-B-Si high strength nickel braze alloys. Examples include - BNi-1 (AMS 4775); BNi-2 (AMS 4777); BNi-4 (AMS 4779). The elements boron and silicon serve as melting point suppressants, lowering the braze alloy melting point substantially below that of the component to be repaired. Unfortunately, the elements boron and silicon are also responsible for the poor ductility of joints brazed with conventional alloys. Linear boride and silicide chains are formed throughout the braze joint. These chains are hard and brittle by nature. With regard to repairing of cracks, the brittle linear chains form along the braze joint providing a ready path for the crack to retrace.

During the seventies, braze alloy development focused on elimination of these brittle boride and silicide chains. It was known that boron could be forced to migrate (diffuse) into the base material by exposing the joint to a post-braze diffusion cycle. Silicon, on the other hand, is confined to the braze joint. Therefore, substitutions for silicon as melting point suppressants were evaluated. Elements tantalum, yttrium and lanthanum were found to be suitable substitutes. The rare earths yttrium and lanthanum assist in the suppression of boride chain formation and contribute to structural spheroidization. A new series of braze alloys was developed on these principles. The alloys were silicon-free and the boron would be forced to migrate into the base material during a post-braze diffusion cycle. Hence, the name "Diffusion Braze Alloys".

4. DIFFUSION BRAZING OF COBALT BASE ALLOY COMPONENTS:

Braze tests were conducted on cobalt base alloy AMS 5382, commercially known as X-40 or Stellite 31. This alloy is used extensively for casting N.G. Vanes that operate at moderate temperatures. The evaluation program is comprised of the following five (5) phases:

I Braze Alloy Flow Testing
II Brazing of Narrow Gap Defects
III Brazing of Wide Gap Defects
IV Stress Rupture Testing of Brazed Bars
V Braze Repairing Sample N.G. Vanes.

4. DIFFUSION BRAZING OF COBALT BASE ALLOY COMPONENTS: (Cont'd)

Successful braze joints are dependent upon the pre-braze joint cleanliness and the as-brazed properties of the braze joint. Pre-cleaning processes available may be categorized as either mechanical or non-mechanical. Hand-held pneumatic grinders and rotary belt sanders are typically the tools used to mechanically rid braze surfaces of contaminants. Non-mechanical cleaning processes rely on chemical reactions to free brazing surfaces of unwanted contaminants. A very effective non-mechanical method of pre-cleaning cobalt base alloys is referred to as hydrogen reduction. The components to be cleaned are exposed to a hydrogen rich atmosphere at elevated temperatures. A typical cycle is 2100°F. for two (2) hours. This method is superior to mechanical cleaning processes for several reasons; mechanical cleaning is limited to preparing surfaces that are accessible with hand tools only; excessive amounts of sound base material are removed when preparing deep/narrow cracks.

Initial braze tests were conducted with six (6) separate braze alloys; four (4) nickel base diffusion braze alloys (VA-N1 through VA-N4); two (2) cobalt base diffusion braze alloys (VA-C1 and VA-C2); one (1) conventional Ni-Cr-B-Si alloy (AMS 4775). The conventional alloy is commonly specified by the O.E.M.'s for repair of both nickel and cobalt base N.G. Vanes. Data obtained from testing this braze alloy would serve as a reference for the diffusion braze alloys.

PHASE I, ALLOY FLOW CHARACTERISTICS:

Specimens were prepared, brazed and metallographically examined in an effort to optimize the braze cycles for each of the braze alloys. Specimen configuration was as shown in Figure 3.

Suitable cycles were soon found for all but the cobalt base diffusion braze alloys. Back-to-back testing at the same braze parameters were plagued by inconsistent results in braze alloy flow characteristics. The braze alloy supplier was contacted and it was concluded this was an inherent problem with the alloy blends. As a result of the discussion, the cobalt diffusion braze alloys were excluded from further testing.

PHASE II, BRAZING OF NARROW GAPS:

Specimens were prepared to simulate the ability of the braze alloys to repair thermal fatigue cracks; deep, narrow width braze joints. Thermal fatigue cracking is the primary cause for rejection of N.G. Vanes from service. Therefore, emphasis was placed on this phase of the evaluation program.

Specimen configuration was as utilized for flow testing (Figure 3), effectively simulating the aspect ratio of thermal fatigue cracks.

Prior to performing the Phase II braze tests, a hydrogen reduction effectiveness test program was performed. The purpose of the program was to verify the effectiveness of hydrogen reduction cleaning. Two (2) batches of braze specimens were prepared; Batch "A" were specimens with mechanically ground braze surfaces; Batch "B" specimen braze surfaces were mechanically ground, oxidized and hydrogen reduced. After assembling and alloying (Figure 3), all specimens were brazed during the same furnace cycle. The subsequent metallographic examination did not reveal any differences in braze flow characteristics (Ref: Figures 4 and 5).

PHASE II, BRAZING OF NARROW GAPS: (Cont'd)

As a result, it was concluded that reduction by hydrogen rich atmospheres could sufficiently eliminate oxides created during in-service operations.

For reference purposes, braze tests with the conventional alloy, AMS 4775, were performed first. The resulting braze joints displayed exceptional wetting and flow characteristics. Porosity was minimal, a characteristic none of the diffusion braze alloys could match. Magnification of the joint at 400X (Figure 6) reveals that the joint is comprised of the following three (3) distinct regions:

1. Transition Region - Diffusion zone at substrate (dendritic cast microstructure) to braze alloy interface.
2. Nickel Rich Region - Light grey zone that separates linear chains from diffusion zone.
3. Intermetallic Region - Centre region of braze joint that displays linear formation of intermetallics.

This regional separation was not evident in diffusion braze alloy joints. However, the levels of porosity were of concern. Provided is Figure 7 which is a typical microstructure of the nickel base diffusion braze alloys. Note the homogeneous distribution of intermetallics throughout the braze joint.

The effect of the diffusion treatment that is responsible for diffusion of the boride and increasing the strength of the joint could not be detected using conventional metallographic techniques. Samples of diffused and non-diffused braze joints were forwarded to the National Research Council (N.R.C.) in Ottawa for evaluation. N.R.C. is presently investigating microprobe analysis as a boron diffusion evaluation technique.

As the levels of porosity in the narrow gap braze specimens were of concern, braze tests were performed on 304 stainless steel sheet. The purpose was to eliminate casting porosity of the AMS 5382 specimens as the cause or contributor of the braze joint porosity. The resulting microstructures displayed varying amounts of porosity. It was concluded that porosity was an inherent characteristic of the diffusion braze alloys.

PHASE III, BRAZING OF WIDE GAPS:

The defects simulated in this phase of the program would be a result of turbine foreign object damage (F.O.D.), severe erosion, or large pieces of base material lost due to connection of cracks.

To repair large defects, diffusion braze alloys are blended with varying ratios of base material powder. The purpose is to provide a joint of "casting" with properties similar to that of the component base material.

The design of the specimens utilized for wide gap brazing is provided in Figure 8. Optimal results were obtained with a blend of braze alloy to base material powder of 40:60. Figures 9 and 10 are typical cross-sections of the resulting braze repair joints.

PHASE III, BRAZING OF WIDE GAPS: (Cont'd)

Modification of the diffusion cycle made no noticeable differences to the braze joint until the diffusion temperature reached the braze joint remelt temperature. However, this is not the purpose of the diffusion cycle. Further investigation was therefore halted.

PHASE IV, STRESS RUPTURE TESTING:

Stress rupture testing was performed in accordance with base alloy specification AMS 5382. The basic requirements are as follows:

Test Temperature	1500 ± 3°F.
Test Load	30,000 psi
Minimum Rupture Time	15 hours
Minimum Elongation	6 percent

The fabrication steps necessary to produce the stress rupture bars are shown in Figure 11. All specimens were x-ray inspected before and after fabrication to ensure integrity of the base material.

Results of the stress rupture tests are provided in Tables II and III. Results from specimens that were post-braze hot isostatic press (H.I.P.) processed are included in the Tables. This process step was added in an attempt to eliminate the porosity of the braze joints. The general effect was to improve the elongation values while the stress rupture life remained unaffected. Note that several specimens failed in the base material, not the braze joint.

Stress rupture data for specimens brazed with alloy AMS 4775 are not included in the Tables as the specimens failed during the after-braze grinding operation. We believe the AMS 4775 braze joints would have displayed adequate stress rupture properties. However, their failure only proved to support the fact that inherent to this series of alloys is poor ductility.

PHASE V, COMPONENT REPAIR:

The object of this phase was to trial repair actual service damaged turbine components using the experience gained in the previous phases of the program. Trial repairs were conducted on first stage nozzle guide vanes from an Allison T56/501 Series III engine. The units are cast from cobalt superalloy AMS 5382 (Stellite 31/X-40). A sample component is shown in Figure 12. A synopsis of the overhaul process is as follows:

1. Incoming inspection.
2. Dissassembly - removal of end caps
   - removal of air distribution tubes.
3. Pre-Inspection Cleaning - stripping of coating
   - abrasive blasting.
4. Inspection - visual
   - dimensional
   - liquid penetrant inspection
   - documentation of damage locations.
5. Pre-braze Cleaning - vapour degrease
   - hydrogen reduction.

PHASE V, COMPONENT REPAIR: (Cont'd)

6. Brazing - apply diffusion braze alloy
   - vacuum furnace braze.
7. Post-braze inspection.
8. Vacuum diffusion cycle.
9. Bench dressing braze areas.
10. Post dressing inspection - visual
    - dimensional
    - liquid penetrant inspection.
11. Assembly - brazing of air distribution tubes and end caps.
12. Coating - application of diffusion coating.
13. Final Inspection - visual.

The detailed inspection revealed that thermal fatigue cracking was the primary cause for retiring the units from service. The airfoil trailing edges suffered the most damage.

Initially, the components were processed per steps 1-10 only noted above. After completion of step 10, the components were sectioned and the repair areas were evaluated metallographically. The results were very encouraging for all the diffusion braze alloys. The amount of joint porosity was far less than that experienced during the earlier stages of the program. A typical repair area cross-section is provided in Figure 13.

With satisfactory results obtained from the trial repairs, several units were then processed per steps 1-13 noted above. The main purpose was to determine if any problems would be experienced during the application of the component protective coating. An aluminum chromium diffusion coating was applied. During visual examination, there were no discernible differences in coating texture or colour between the repair and non-repair areas. Metallurgical examination disclosed the diffusion of the coating into the braze alloy rich areas (Figure 14).

5. CLEANING OF γ' STRENGTHENED SUPERALLOYS:

As noted earlier, the inherent feature of the high temperature high strength nickel base superalloys is a precipitation referred to as γ' [$Ni_3(Al,Ti)$]. The compatability of crystal structure and lattice constant between the nickel matrix and the precipitate allows homogeneous nucleation with low surface energy. The result is a long time stability of the alloy and remarkably, the strength of γ' increases as temperature increases (below 0.8 Tm).

It is ironic that the alloying elements of aluminum or titanium which most significantly contribute to the exceptional performance of these alloys render them almost unrepairable. The problem arises with aluminum and titanium's strong affinity for oxygen. The oxides formed are complex spinels. Such spinels are not formed on cobalt base superalloys. The complex spinels have a low free energy value and hydrogen rich atmospheres lack the potential to reduce the oxides. A successful braze cannot be performed on an oxidized surface.

Currently, only cleaning processes using active gaseous fluoride ions are successful in reducing the aluminum and titanium oxides. There are several organizations that perform fluoride ion reduction in Europe and North America. However, only recently has it been adopted to processing service exposed turbine components.

5. CLEANING OF γ' STRENGTHENED SUPERALLOYS: (Cont'd)

Several facilities rely on the thermal decomposition of salts or fluorocarbon gases. A weak point of this technique is the lack of precise control over the concentration of fluoride ions once decomposition has started. An alternate technique which relies on the controlled injection of hydrogen fluoride gas into a cleaning chamber shows more promise. Due to modern mass flow meter technology, the concentration of hydrogen fluoride can be precisely controlled during processing. The end result is a more consistent cleaning technique. A schematic of a hydrogen fluoride injection cleaning system is shown in Figure 15.

Exposure of γ' alloys to the hydrogen fluoride (HF) process allows the following reactions to occur:

1. The HF gases react with aluminum and titanium at the alloy surface to form their fluorides:

$$12\ HF\ (g) + 2\ Al_2O_3 \xrightarrow{T > 700°C.} 6\ H_2O\ (g) + 4\ AlF_3$$

2. Surface depletion of aluminum and titanium occurs:

$$3\ HF + Ti \longrightarrow Ti\ F3 + 3/2\ H_2$$

This secondary reaction is of great benefit as it prevents the reformation of aluminum and titanium oxides at the alloy surface during the subsequent braze repair cycle.

6. FUTURE ACTIVITIES:

Service evaluation is the final and often most important phase in turbine component repair development. With regard to N.G. Vanes, the two (2) test options available are: in-service testing and burner rig testing. As nozzle guide vanes are non-rotating components and they are often removed from service with substantial damage, in-service testing of these components is recommended. The proposed test vehicles for repair of cobalt base N.G. Vanes are Allison T56 Series engines operated by the Canadian Department of National Defence (D.N.D.). An approved on-wing evaluation program is targeted for the fourth quarter of 1987.

In conjunction with the National Research Council of Canada, a program was devised to develop diffusion braze repair techniques for nickel base γ' strengthened N.G. Vanes. The program parallels that of the cobalt base alloy evaluation program with the following exceptions:

1. A complete active fluoride oxide reduction (AFOR) system has been constructed at the Vac-Aero International, Oakville facility.

2. High temperature low cycle fatigue testing of braze joint specimens has been added to the program.

The tentative completion date for the nickel base alloy program is October of 1988.

REFERENCES:

Decker, R.F., The International Nickel Company, Inc., Strengthening Mechanisms In Nickel-Base Superalloys, 1969.

Hermanek, F.J., Stern, M.J., Alloy Metals, Inc., Turbine Component Restoration By Activated Diffusion Brazing, 1984, Ref. No. 8407-004.

Haafkens, M.H., Matthey, J.H.G., Interturbine Holland, A New Approach To The Weldability Of Nickel-Base As-Cast And Powder Metallurgy Superalloys.

Haafkens, M.H., Experiences In Repair Of Hot Section Gas Turbine Components, SAE Technical Paper Series, Ref. No. 821490, 1982.

Adam, P. Dr., Steinhauser, L., MTU-München, Bonding Of Superalloys By Diffusion Welding And Diffusion Brazing, AGARD Conference Proceedings No. 398, 1985.

Mom, A.J.A. (1), Madhava, N.M. (2), Kool, G.A. (3), Dean, M., (1) National Aerospace Laboratory, Amsterdam, The Netherlands, (2) Chromalloy Division Oklahoma, U.S.A., (3) Turbine Support Europa, The Netherlands, Evaluation of DDH and Weld Repaired F100 Turbine Vanes Under Simulated Service Conditions, AGARD Conference Proceedings No. 398, 1985.

Mattheij, J.H.G., Elbar BV, Lomm, The Netherlands, Role Of Brazing In Repair Of Superalloy Components - Advantages And Limitations, presented at The Metals Society, London, on May 16, 1984.

Bell, S.R., Turbine Services Ltd., Hatfield, Hertfordshire, Repair And Rejuvenation Procedures For Aero Gas-Turbine Hot-Section Components, presented at The Metals Society, London, on May 16, 1984.

Brauny, P., Hammerschmidt M., Malik, M., Deutsche Lufthansa AG., Hamburg, Federal Republic of Germany, Repair of Air-Cooled Turbine Vanes of High-Performance Aircraft Engines - Problems And Experience, presented at The Metals Society, London, on May 16, 1984.

Armstrong, R.J., Vac-Aero International Inc., Oakville, Ontario, Canada, Activated Diffusion Braze Alloy Test Program, January 15, 1987.

ACKNOWLEDGEMENTS:

With pleasure the authors acknowledge the assistance of J.P. Immarigeon of the National Research Council of Canada, Ottawa and L.C. Dyke of Standard Aero Limited, Winnipeg, Manitoba, Canada.

TABLE I

NOZZLE GUIDE VANE ALLOYS

(Composition In Percentage By Weight)

COBALT BASE

	Ni	Co	Cr	C	Mo	W	Fe	Al	Ti
AMS 5382 (X-40)	10.5	BAL	25.5	0.50	0.50	7.5	2.0	0	0

NICKEL BASE

	Ni	Co	Cr	C	Mo	W	Fe	Al	Ti
RENE 80	BAL	9.5	14	0.17	4.0	4.0	0	3.0	5.0
MAR-M246	BAL	10.0	9.0	0.15	2.5	10.0	0.15	5.5	1.5
IN-738	BAL	8.5	16.0	0.17	1.75	2.6	0.50	3.4	3.4
IN-100	BAL	15.0	10.0	0.18	3.0	0	0	5.5	4.7

T A B L E II

STRESS RUPTURE VALUES

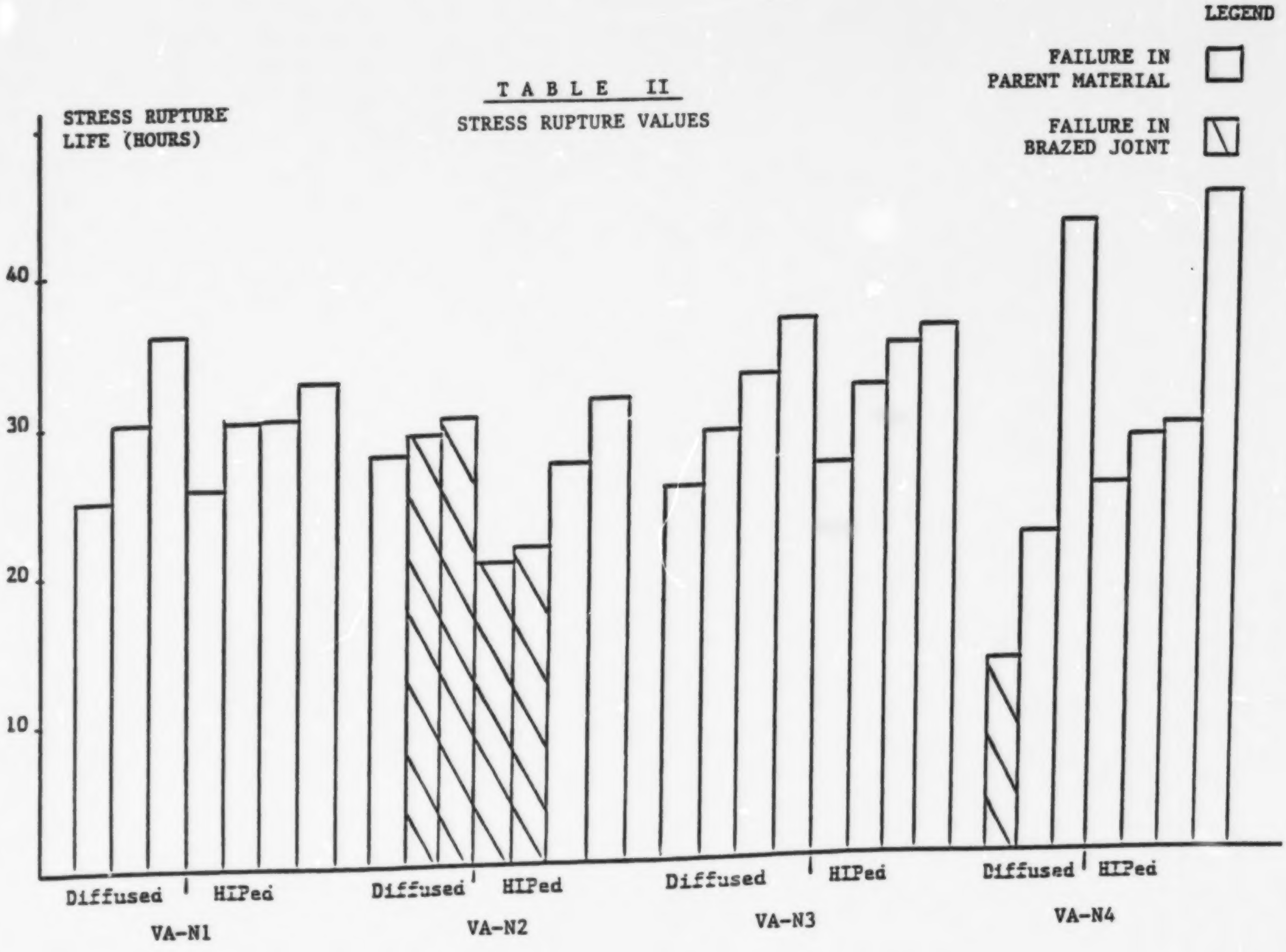
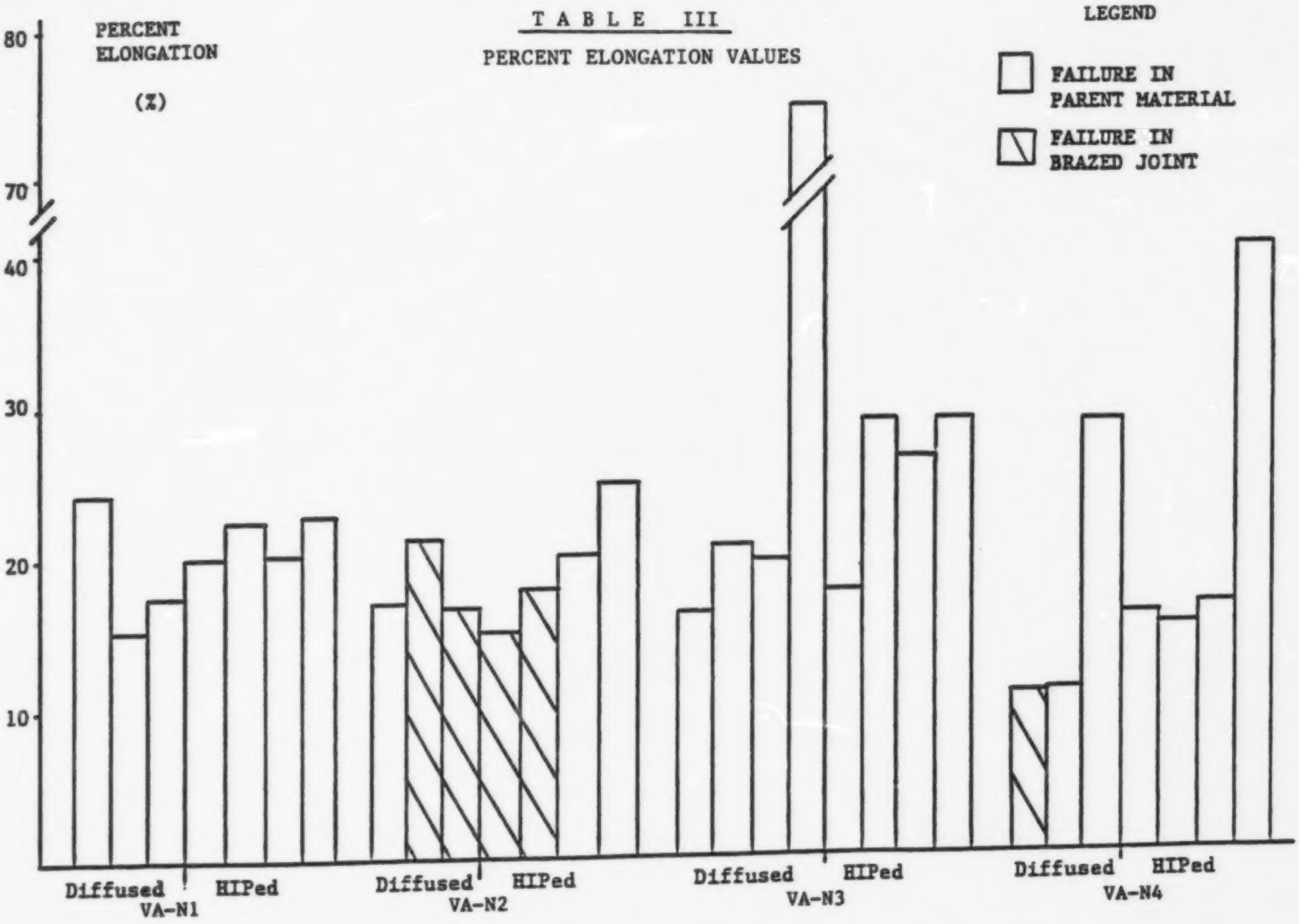


TABLE III

PERCENT ELONGATION VALUES



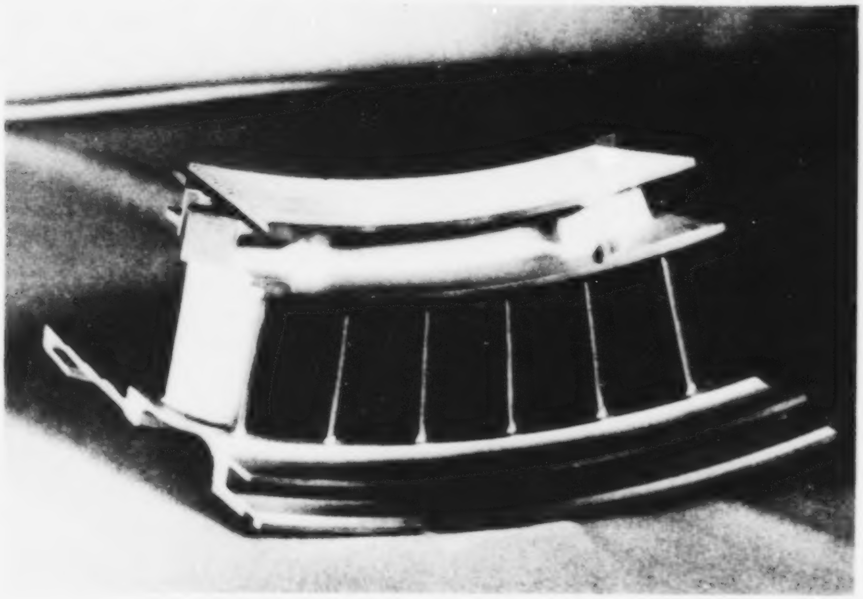


FIGURE 1
FIRST-GENERATION N.G. VANE

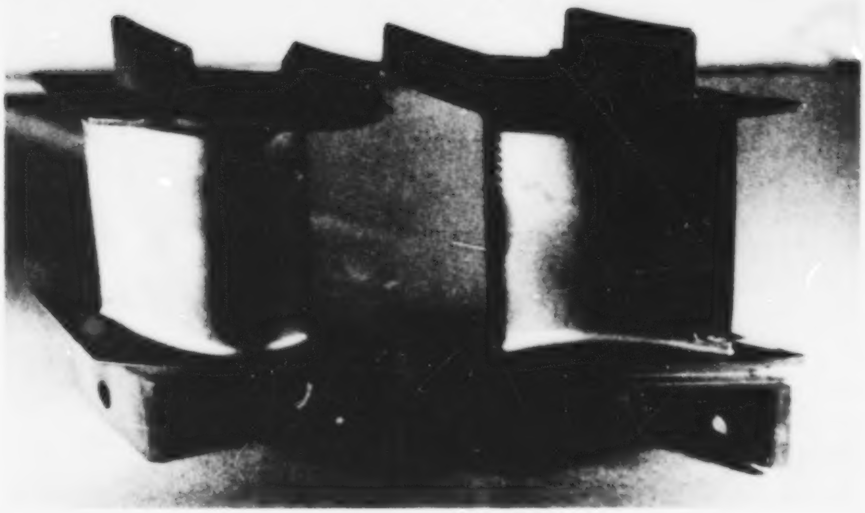


FIGURE 2
SECOND-GENERATION N.G. VANE

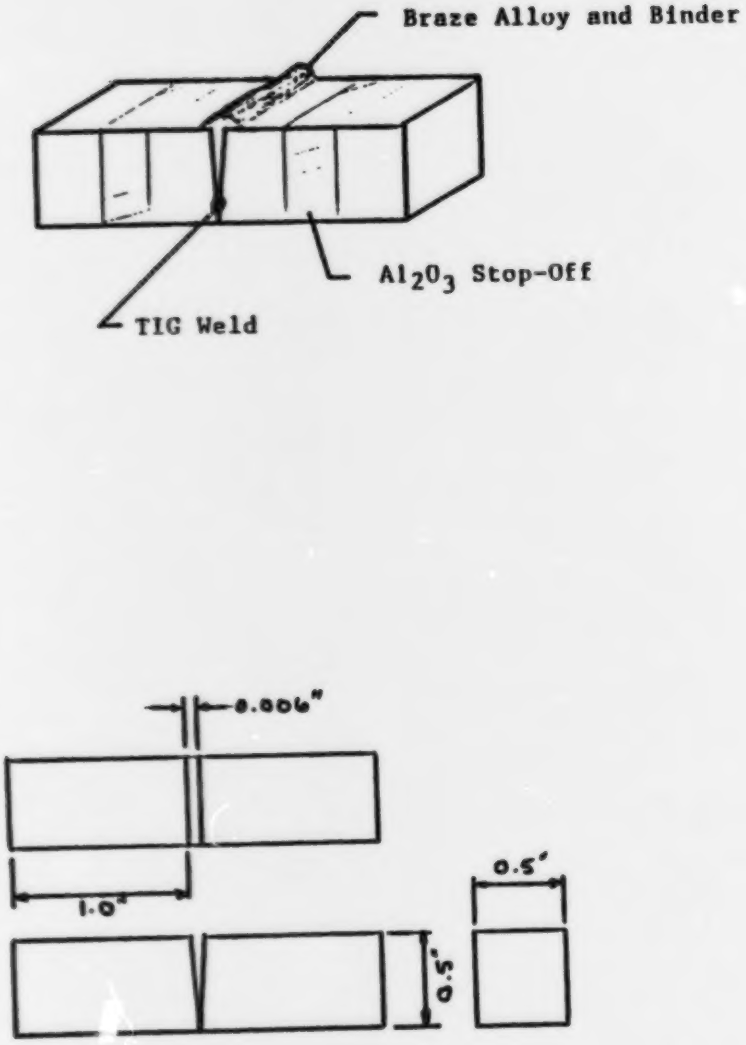

Braze Alloy and Binder
Al2O3 Stop-Off
TIG Weld
0.006"
1.0"
0.5'
0.5"

FIGURE 3
NARROW GAP BRAZE TEST SPECIMEN CONFIGURATION

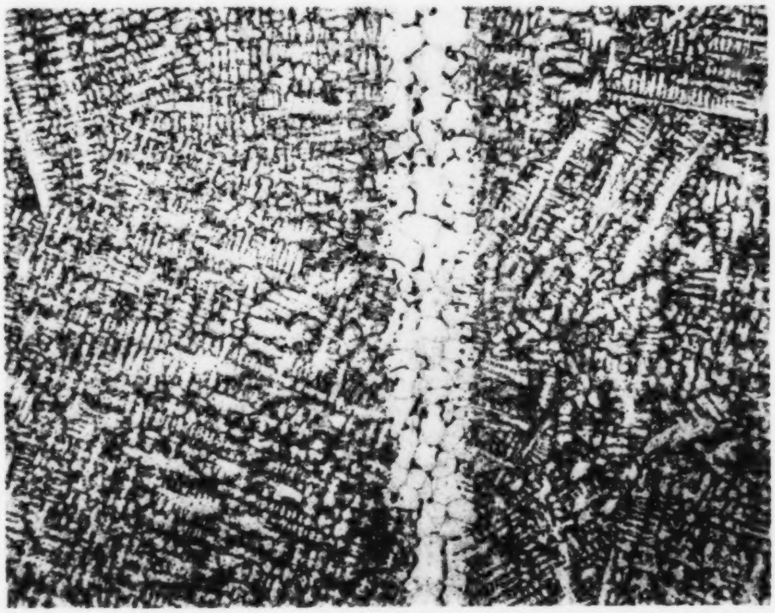


FIGURE 4
AS-GROUND VA-N3 BRAZE JOINT
50X $KMnO_4$

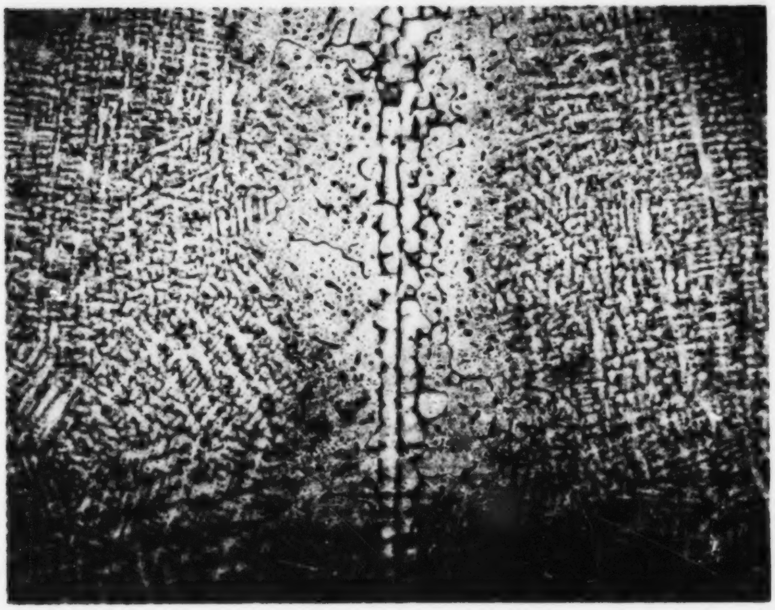


FIGURE 5
OXIDIZED, HYDROGEN REDUCED BRAZE JOINT
50X $KMnO_4$

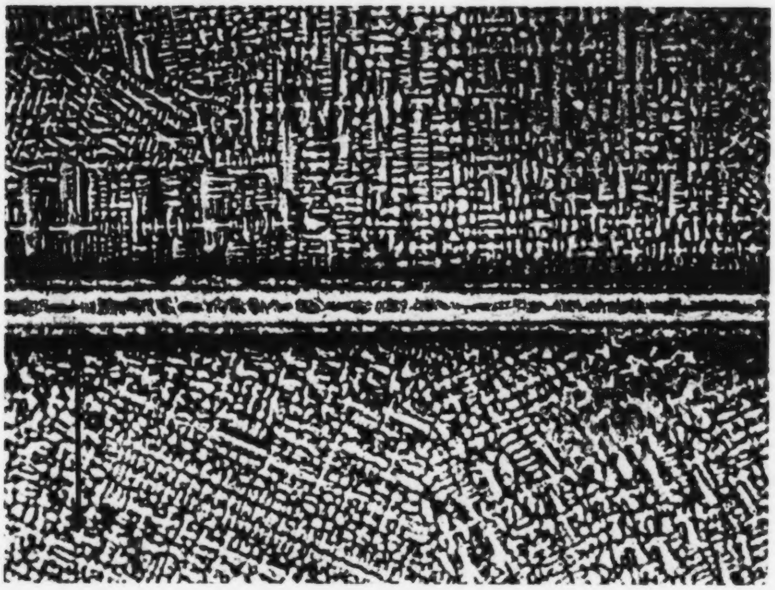


FIGURE 6
AMS 4775 TEST JOINT

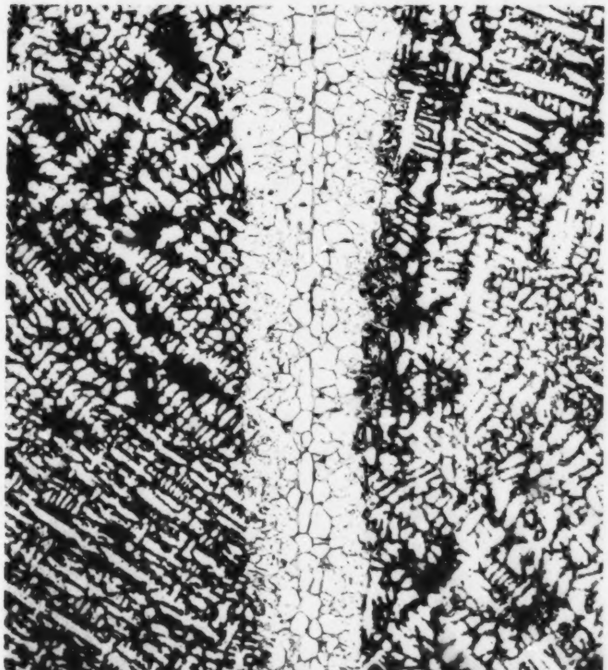


FIGURE 7
DIFFUSION BRAZE ALLOY VA-N1
50X ELECTROLYTIC OXALIC

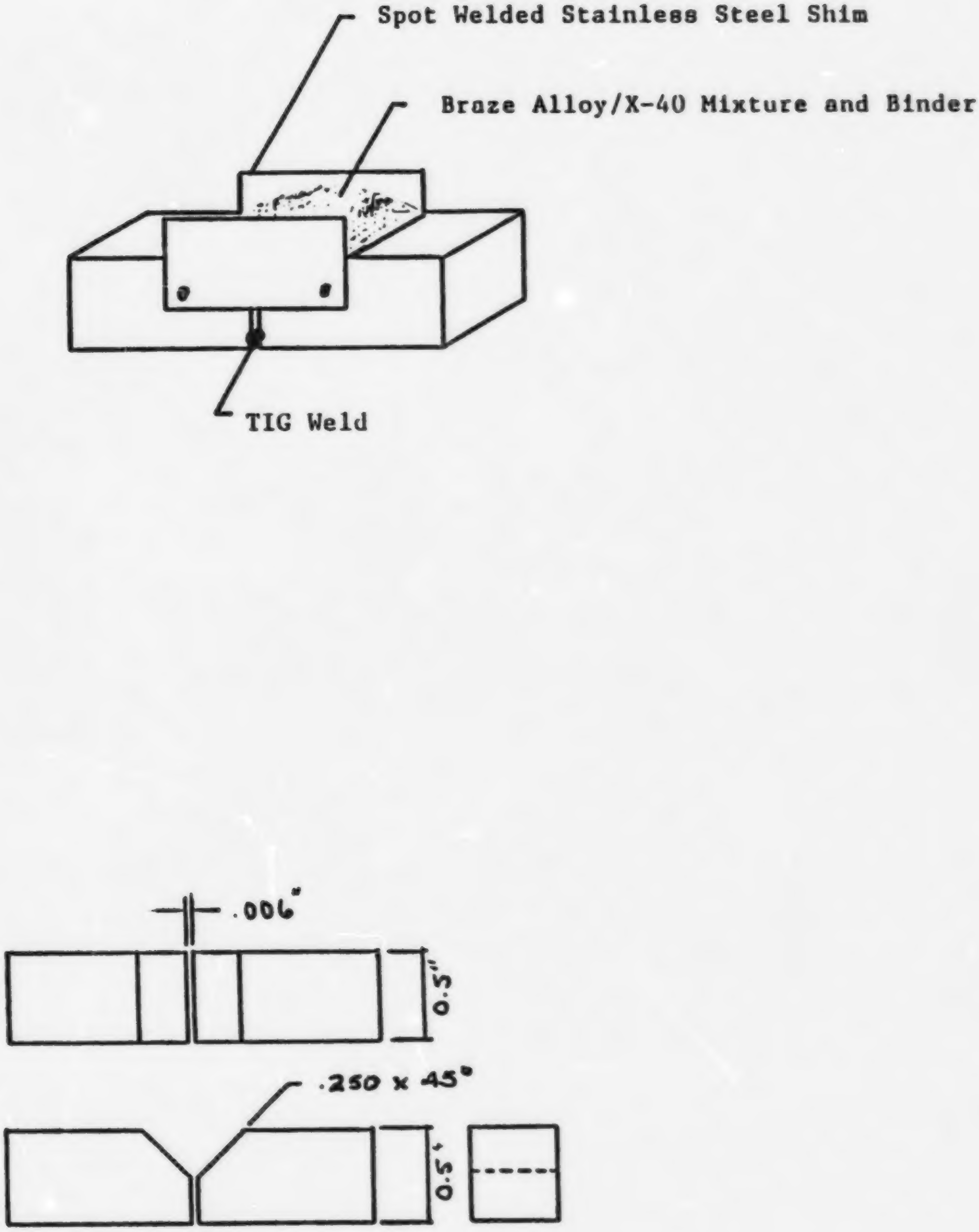

Spot Welded Stainless Steel Shim
Braze Alloy/X-40 Mixture and Binder
TIG Weld
.006"
0.5"
.250 x 45°
0.5"

FIGURE 8
WIDE GAP TEST SPECIMEN CONFIGURATION

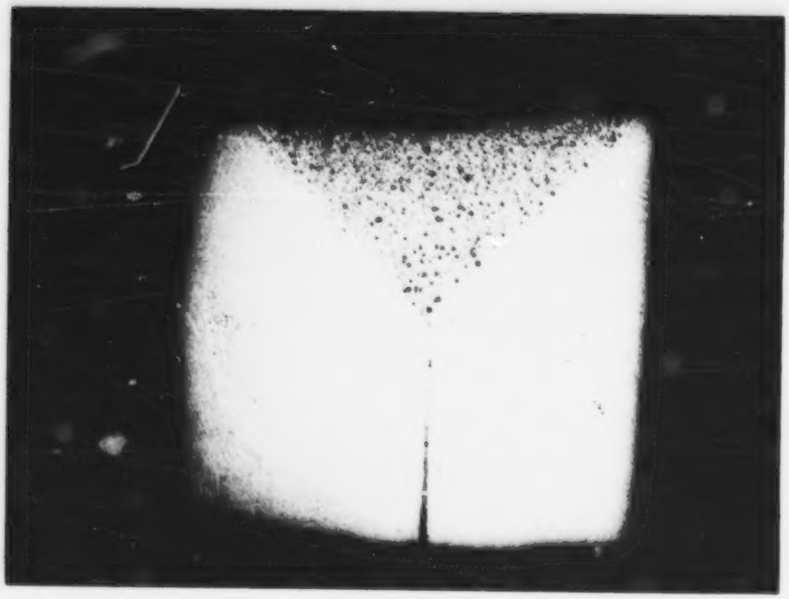


FIGURE 9
WIDE GAP JOINT BRAZED WITH VA-N2
50X $KMnO_4$

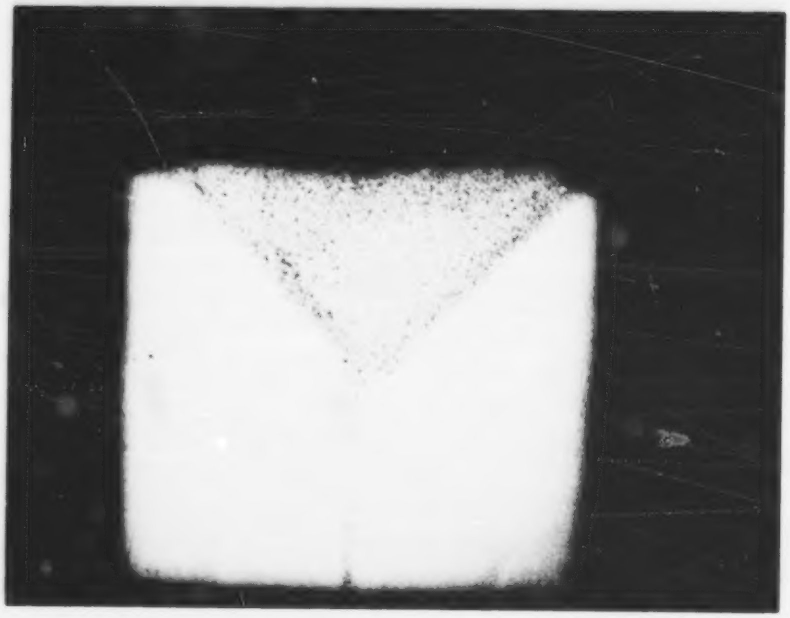
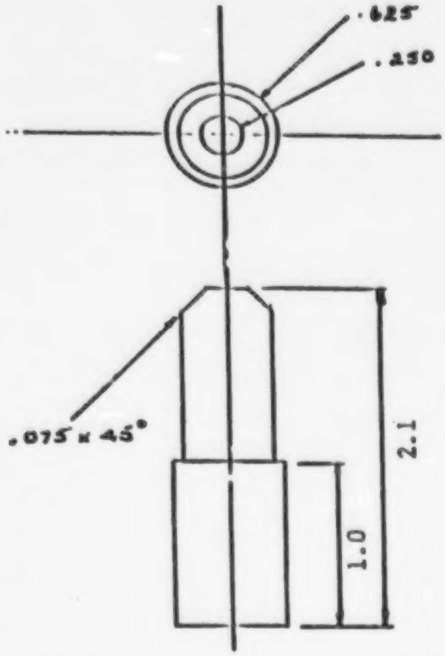
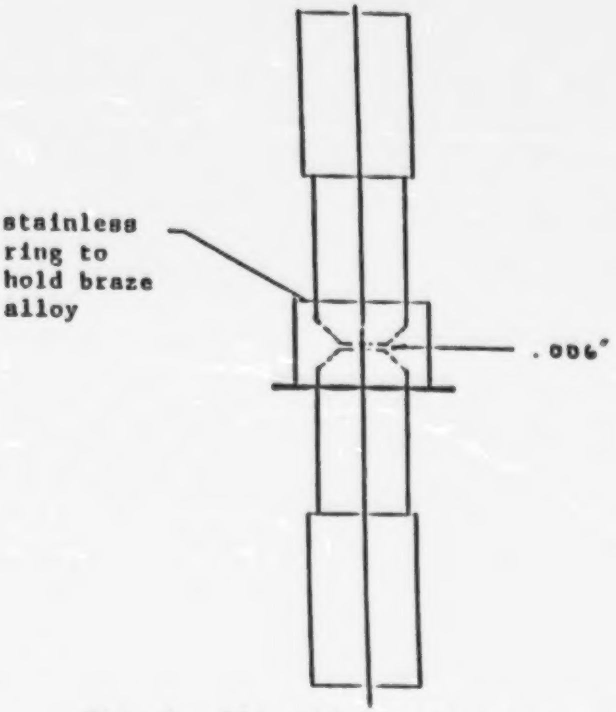


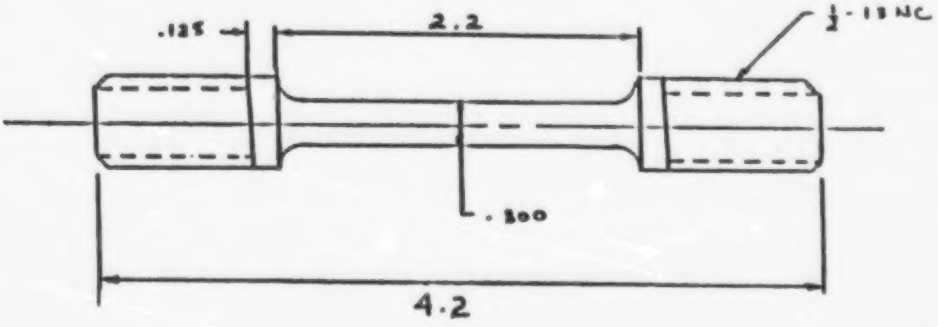
FIGURE 10
WIDE GAP JOINT BRAZED WITH VA-N3
50X $KMnO_4$


.625
.250
.075 x 45°
2.1
1.0

Step 1: Febrication of ½ of pre-braze stress rupture bar.


stainless ring to hold braze alloy
.006"

Step 2: TIG weld two halves to produce stress rupture bar


.125
2.2
½-13NC
.300
4.2

Step 3: Machined stress rupture test sample.

FIGURE 11
FABRICATION OF STRESS RUPTURE SPECIMENS

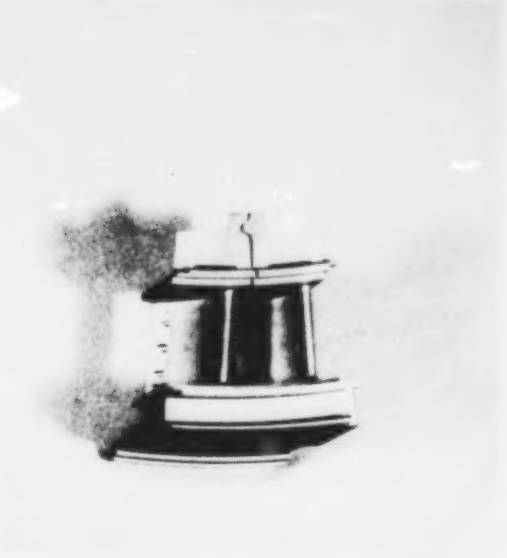


FIGURE 12: ALLISON T56/501 FIRST STAGE N.G. VANE

FIGURE 13: TYPICAL REPAIR AREA, T56/501 N.G. VANE, ALLOY VA-N1 50X OXALIC ACID

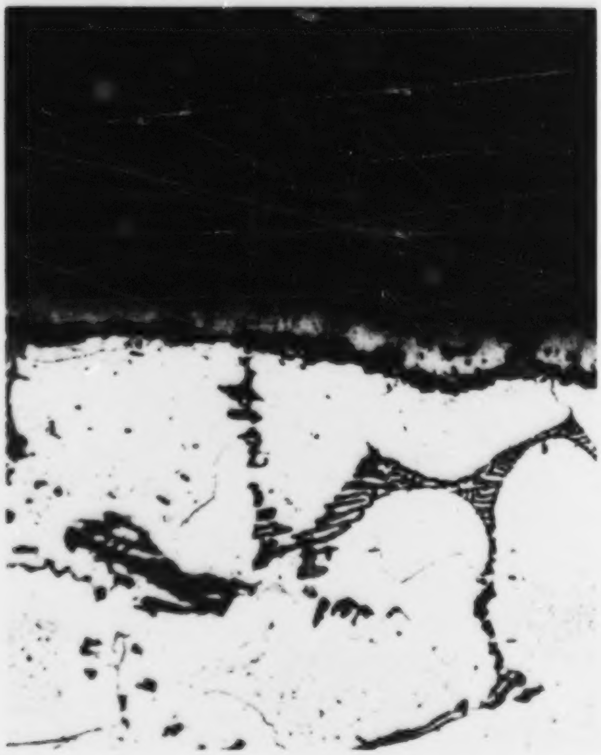


FIGURE 14: ALUMINUM CHROMIUM DIFFUSION COATING ON VA-N1 DIFFUSION BRAZE 400X ELECTROLYTIC OXALIC

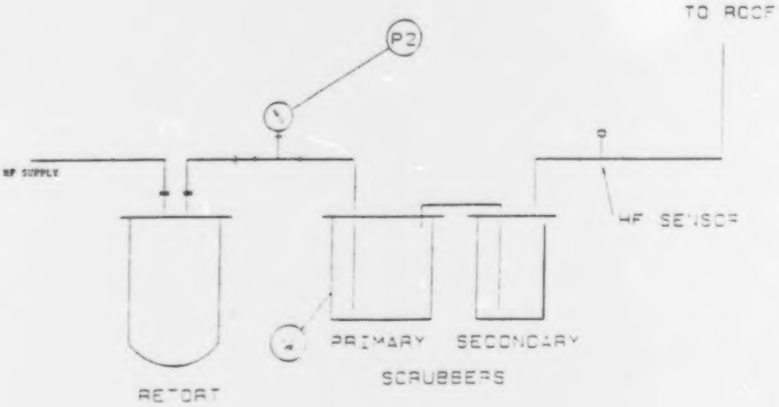

TO ROOF
P2
HF SUPPLY
HF SENSOR
PRIMARY
SECONDARY
SCRUBBERS
RETORT

FIGURE 15: SCHEMATIC OF HF INJECTION CLEANING SYSTEM

OPERATION OF OIL-LESS SEAL AND BEARING SYSTEMS FOR CENTRIFUGAL COMPRESSORS

ROEL P. LANCEE, P. ENG.

Senior Engineer
Station Design
TransCanada PipeLines Limited
P.O. Box 54, Commerce Court West
Toronto, Ontario, Canada
M5L 1C2

ABSTRACT

Booster compressors on a natural gas pipeline system have long been plagued by the problems associated with oil entrainment in the natural gas stream and the inherent maintenance problems associated with oil systems. The use of dry gas seals and active magnetic bearings will effectively eliminate the further use of oil systems in gas compression.

The paper will deal with the history of TransCanada PipeLines' past experiences with oil-free seals, the theory of dry gas seals and active magnetic bearings, the effects on rotor dynamics by magnetic bearings, and the recent installation and operation of a set of dry gas seals and active magnetic bearings in a booster compressor unit operating on the TransCanada PipeLines system.

CONTENTS

ABSTRACT

1.0 INTRODUCTION

2.0 DRY GAS SEALS

3.0 ACTIVE MAGNETIC BEARINGS
    3.1 Magnetic Bearings and Rotor Dynamics

4.0 ELECTRONIC CONTROL LOOP

5.0 THE INSTALLATION

6.0 TUNING

7.0 OPERATING PROCEDURES
    7.1 Dry Gas Seals
    7.2 Magnetic Bearings

8.0 PERFORMANCE
    8.1 Control Loop
    8.2 Dry Gas Seals

9.0 ECONOMIC JUSTIFICATION
    9.1 Dry Gas Seals
    9.2 Active Magnetic Bearings

10.0 CONCLUSIONS

REFERENCE

1.0 INTRODUCTION

The conventional oil film shaft seals and tilt pad journal bearings are presently used by turbocompressor manufacturers. Their technology and applications have proven most reliable in the past, however, today the compression industry is striving to operate its units and associated pipeline systems in a more efficient manner. Oil film seals unfortunately are designed to allow minor leakages into the gas stream and subsequently into the pipeline system thereby reducing the overall system's efficiency.

Technological advancements in the form of active magnetic bearings and dry gas seals in an oil-free environment have recently been introduced into the field of industrial machinery. These advancements promise the user lower capital, operating and maintenance costs for both the retrofit of existing equipment and for new compression and eventually the total elimination of oil systems.

2.0 DRY GAS SEALS

TransCanada PipeLines (TCPL) first considered dry gas seals some ten years ago with very little success. These earlier designs depended upon hydrostatic gas forces and an opposing spring force to maintain the desired operating clearance between the seal faces. The inability to maintain the proper clearance eventually made it necessary to revert back to the conventional seal oil systems.

By 1982, the dry gas seal technology had advanced to the stage that seal designs included spiral groves or scallops in the rotating faces, which generated hydrodynamic forces between the rotating and floating stationary rings. Together with the hydrostatic and spring forces, a much more reliable operating gap was maintained under all operating conditions. The seal provided in this TCPL installation comprised a tandem face type with double circumferential seal rings between the seal and the inboard bearing. The tanden face seal was designed to reduce the high pressure filtered buffer gas to a pressure slightly higher than atmosphere. This prototype seal operated reliably for over one year with gas leakage rates of 5 - 9 SCFM (0.0024 - 0.0042 m^3/s) at discharge pressures up to 1000 psi (6895 kPa). Oil migrating from the bearing lubrication system ultimately contaminated the outer sealing face. A modification to the inner bearing oil supply system minimized but did not eliminate the oil from migrating towards the seal faces.

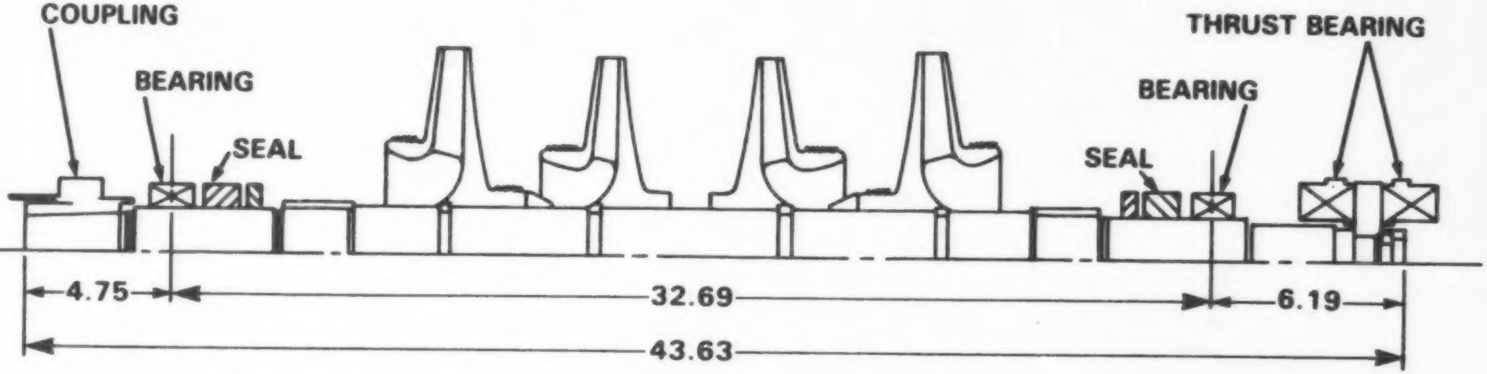
TCPL carried out two more retrofits in 1984. During the operating period 1984 - 85, two out of the three compressors operating with dry seals encountered many of the following problems:

1. Susceptibility of seal cartridge O-rings to damage during installation which caused high gas leakage.

2. Dirty buffer air containing oil in either a liquid or vapour phase passed through the filter, contaminating the seal faces.

3. Ingress of lubricating oil from the inner bearing contaminating the seal faces.

FLUID FILM OIL BEARING/OIL SEAL ARRANGEMENT


COUPLING
BEARING
SEAL
THRUST BEARING
BEARING
SEAL
4.75
32.69
6.19
43.63

MAGNETIC BEARING/GAS SEAL ARRANGEMENT

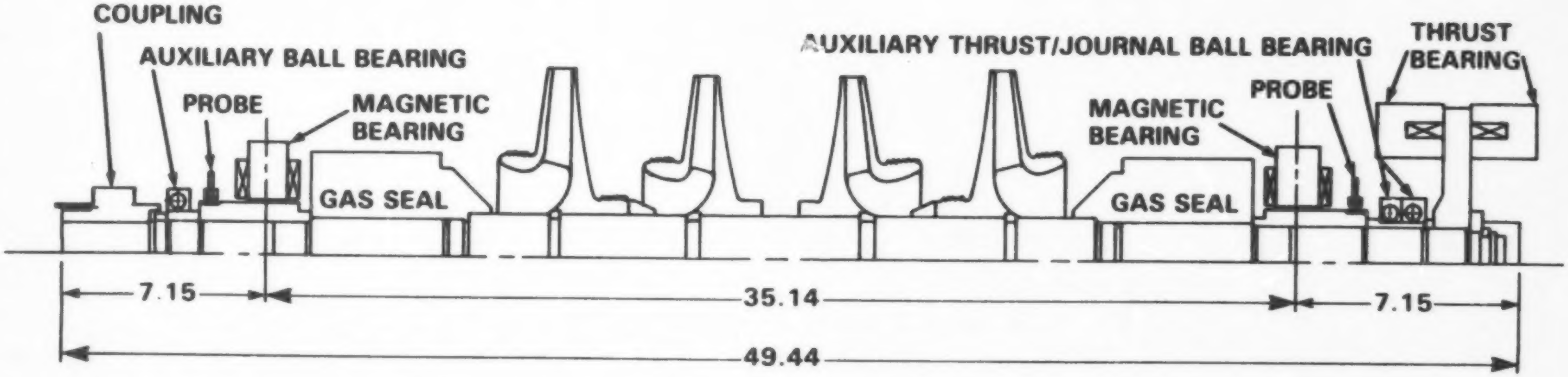

COUPLING
AUXILIARY BALL BEARING
PROBE
MAGNETIC BEARING
GAS SEAL
AUXILIARY THRUST/JOURNAL BALL BEARING
THRUST BEARING
PROBE
MAGNETIC BEARING
GAS SEAL
7.15
35.14
7.15
49.44

Figure 1

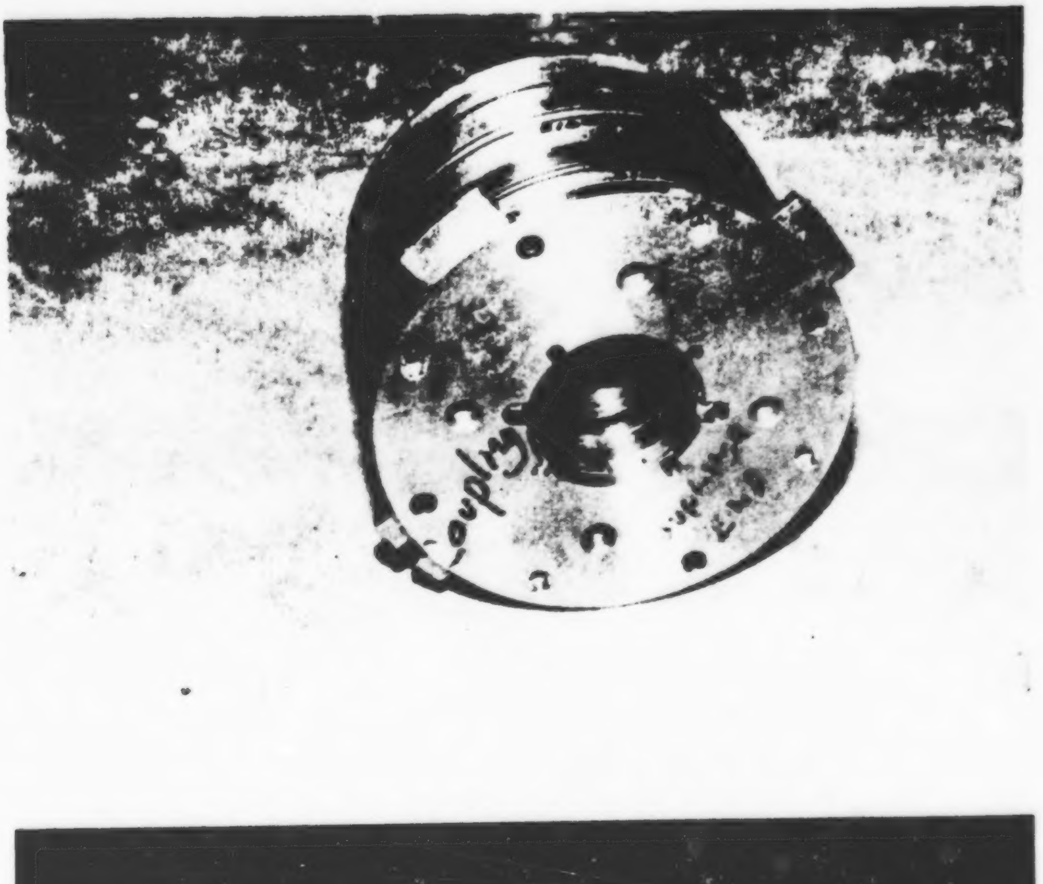


Figure 2
DRY GAS SEAL

4. Conversion to the dry gas seal system caused the compressor thrust bearing to be more heavily loaded on the outboard pads. As a result, there were thrust bearing failures during such upset flow conditions as surge.

5. Suspected unstable rotor dynamics probably caused by misalignment or flexible coupling lockup caused poor seal performance and rapid failure.

These problems have either been resolved or are in the process of being resolved. Based on this limited success and the experience of other users, TCPL felt that the goal of oil free gas seals, although elusive, was realistic.

In November of 1984, TCPL received a proposal to retrofit one of its pipeline compressors with dry gas seals and active magnetic bearings. The seals were to be of a design not yet tried on the TCPL system but proven successful by other users. The active magnetic bearings were yet to be installed in a field unit. The proposal was attractive from the standpoint that it afforded an opportunity to operate and gain confidence in a new seal design and the concept of active magnetic bearings. A backup set of conventional oil film seals and tilt pad journal bearings were readily available should any major difficulties arise. Figure 1 represents a cross-sectional comparison between the subject compressor rotor as originally designed and as retrofitted with magnetic bearings and dry gas seals.

The type of seal used is a mechanical face seal (see Figure 2). The seal is designed with a self-acting geometry which sets up a stable film between the seal faces. The hydrostatic and hydrodynamic forces are balanced to ensure that the film is thick enough to prevent mechanical contact and thin enough to prevent excessive leakage. During operation, the mating ring rotates with the compressor; a spiral grove etched in the mating ring acts as a miniature centrifugal compressor, designed with a high head, low flow characteristic. This compression action creates a highly compressed film of gas which separates the seal faces. The forces acting in the operating seal are such that they balance each other to give a satisfactory operation over the entire operating range of pressures and speeds.

The seal design is unique to the magnetic bearing application. For conventional oil bearings, a carbon ring seal is used, however a gas seal is required because the magnetic bearing is situated in an atmosphere of slightly pressurized air. In addition, the bearing atmosphere (with a controlled leakage) must be sealed from the main seal gas vent. By maintaining a controlled leakage from the bearing area, all the gas seal leakage from the compressor is vented. A positive gas rich mixture is therefore maintained in the vent.

The mechanical seal replaced not only the conventional oil seals and tilt pad journal bearings but also the following auxiliary equipment associated with the seal oil system:

1. Main gear-driven seal oil pump.

2. Auxiliary seal oil pump.

3. Emergency seal oil pump.
4. Gear-driven lube oil pump, (1 only).
5. Oil drainers.
6. Overhead seal oil tank.
7. Miscellaneous valves, pressure switches, level switches and associated piping.

3.0 ACTIVE MAGNETIC BEARINGS

An active magnetic bearing comprises two major, distinct parts; the bearing hardware itself and the electronic control system logic. The bearing contains four distinct components; radial bearings and sensors, axial (thrust) bearing and sensor, speed pick-up, and auxiliary (back-up) bearings.

As in any bearing, the magnetic bearing is essentially composed of the rotor and the stator elements. The rotor consists of a stack of ferromagnetic laminations without slots or windings, force-fitted onto the shaft. The stator is also made up of stacked laminations, but is slotted and includes windings. The technology of the magnetic radial bearing is very close to that of an asynchronous type motor. It is the stator that forms the electromagnetic section of the bearings and the position sensor section (see Figure 3). The electromagnets of the stator are positioned diagonally opposite each other and are designated as the V_1 - V_3, W_1 - W_3, and V_2 - V_4, W_2 - W_4 electrical axes for control of the radial position of the shaft (see Figure 4). The electromagnets of the stator maintain the rotor in magnetic levitation, their magnetic field being modulated as a function of the radial position of the rotor, this being controlled in turn by the position sensors and compared with the desired zero radial position. If the position of the rotor shifts, the electronic control loop, through the position sensor, will detect the displacement. It will then direct current to the appropriate electromagnet in order to pull the rotor back to its nominal position.

The bearings position the shaft not only in the radial direction but also in the axial direction (see Figure 3). The magnetic axial bearing is also designed to use only the magnetic fields of force. Axial control of the shaft is traditionally performed by means of a double acting thrust bearing; in the case of the magnetic bearing, the annular stators, with windings, act on either side of the ferromagnetic thrust collar. They are designated as the Z_1 - Z_2 electrical axis on Figure 4. Axial movement of the shaft is handled in a similar fashion as radial movement using an axially mounted position sensor. The axial sensor need not be located close to the thrust bearing, but in this case was located at the end of the shaft.

Batch-lubricated auxiliary ball bearings are fitted at each end to support the shaft in one half of the magnetic air gap in the stationary, unenergized state or in the event of an electronic system failure.

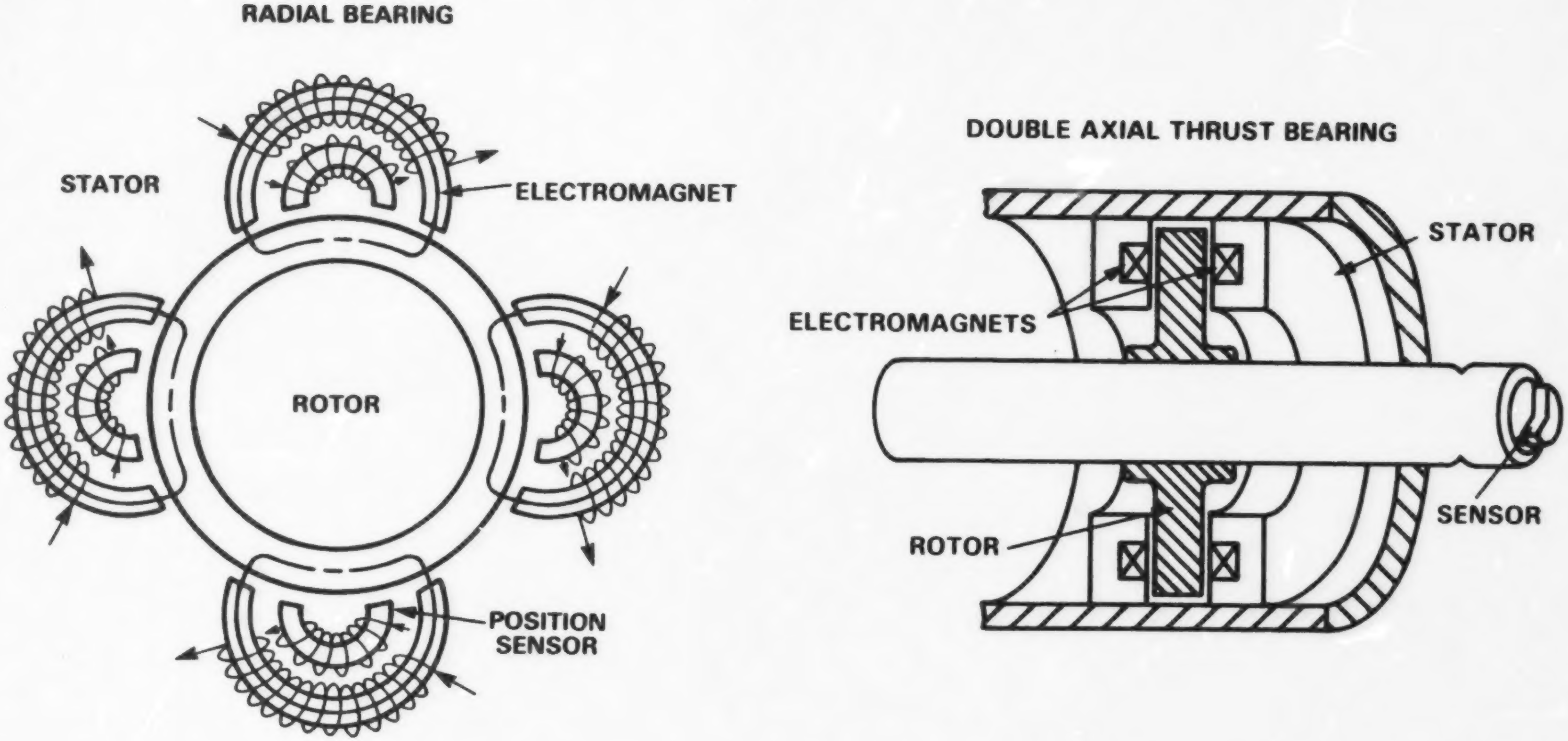
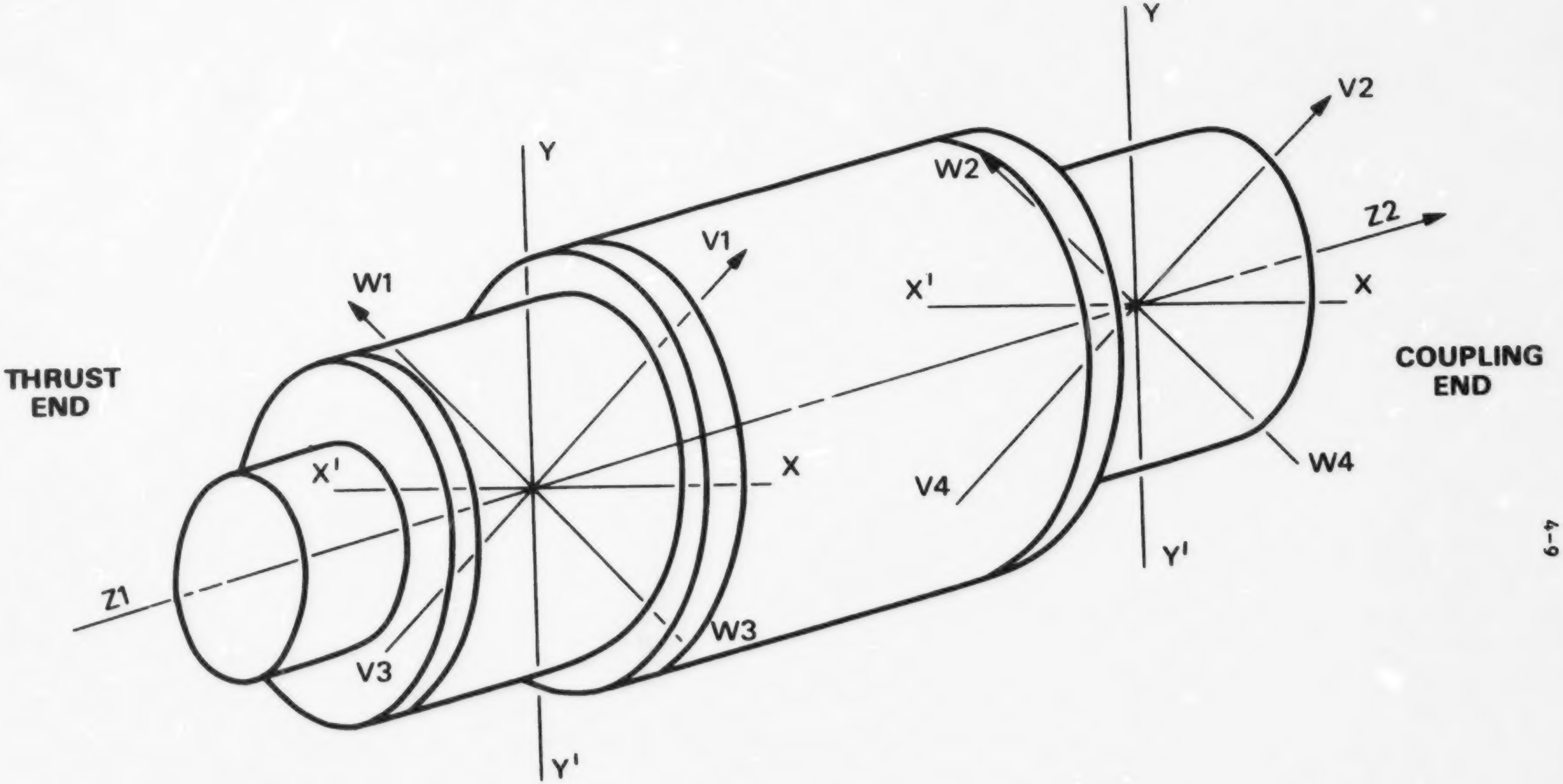

RADIAL BEARING
STATOR
ELECTROMAGNET
ROTOR
POSITION SENSOR
DOUBLE AXIAL THRUST BEARING
STATOR
ELECTROMAGNETS
ROTOR
SENSOR

Figure 3
BEARINGS SCHEMATIC DIAGRAM


THRUST END
COUPLING END
Y
Y'
Y
Y'
X
X'
X
X'
Z1
Z2
V1
V2
V3
V4
W1
W2
W3
W4

X and Y Mechanical Axis

V and W Electric Axis

Figure 4
ORIENTATION OF ELECTRICAL AND MECHANICAL AXES

3.1 Magnetic Bearings and Rotor Dynamics

The application for magnetic bearings provides increased flexibility in the field of rotor dynamics.

The rotor dynamics are no longer subject to the commercially available oil film bearing characteristics. The critical components which result in stable rotor operation such as bearing stiffness and damping are not a limitation; with magnetic bearings, these parameters become variable. Bearing stiffness and damping are functions of magnetic bearing design and calibration. Rather than a rotor designed to conform to the commercially available bearings, the bearing characteristics are designed to accommodate the rotor required for the job. The magnetic bearings, in essence, provide optimum stiffness factors. Table 1 shows the compressor design parameters of the TCPL unit and Figure 5 is representative of the differences in characteristics between the fluid film bearing design and the magnetic bearing design with regard to undamped critical speeds.

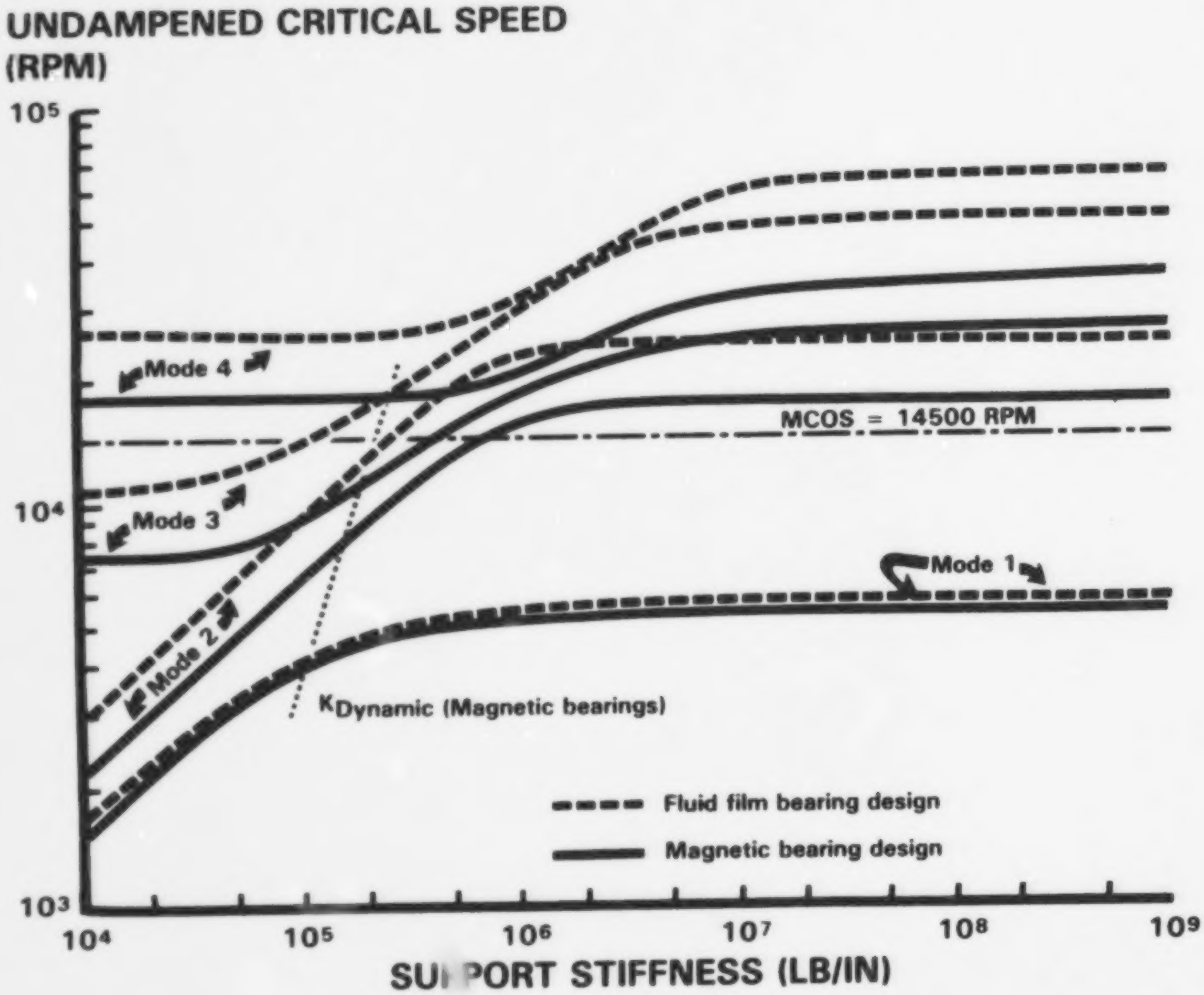


Figure 5

FLUID FILM V MAGNETIC BEARINGS
UNDAMPENED CRITICAL SPEED MAP

TABLE NO. 1

CENTRIFUGAL COMPRESSOR DESIGN PARAMETERS AND NOMENCLATURE

Parameter, Nomenclature	U.S. Units	(SI Units)
Operating Speed, N, RPM (Hz)	14500.	(241.7)
Total Weight, W, lb (N)	278.	(1237)
Bearing Span, in. (mm)	34.88	(886.0)
Shaft Length, in. (mm)	49.32	(1253)
Coupling End Overhand, in. (mm)	7.28	(184.9)
Shaft Stiffness, k_s, lb/in. (N/m)	1.22E5	(21.4)
Bearing Stiffness @ MCOS, k_b, lb/in. (N/m)	1.60E5	(28.0)
Stiffness Ratio, K. Dim., (Dim.)	1.41	(1.41)
Mid-Span Diameter, in. (mm)	3.00	(76.2)
Journal Diameter, in. (mm)	3.66	(93.0)
First Rigid Bearing Critical, N_{cr}, RPM (Hz)	5423.	(90.38)
First Peak Response Speed, FPS1, RPM (Hz)	4300.	(71.67)
Second Peak Response Speed, FPS2, RPM (Hz)	18300.	(305.0)
First Mode Modal Mass, M_m, lb_n (N)	146.	(649)
Bearing Stiffness @ N_1, K_{b1}, lb/in, (N/m)	8.63E4	(15.1)
Optimum Damping (Ref. 9), C_o, lb-sec/in (N-sec/m)	259.	(4.54E-2)

NOTE: Values are given in both SI and U.S. customary units. The measurements and calculations were made in U.S. customary units.

4.0 ELECTRONIC CONTROL LOOP

The active magnetic bearing principle is based on the use of forces created by a magnetic field to levitate the rotor without mechanical contact between the stationary and rotating components. Continuous monitoring of the rotating shaft for any radial and axial displacement is carried out by means of sensors. Signals transmitted from these sensors through the electronic control loop to the four electromagnets, "actively" control the position of the rotor.

The control loop determines the magnitude and direction of the magnetic forces generated. Figure 6 represents a typical control loop block diagram. The relationship between the rotor position and the magnetic force to be generated, is co-ordinated in the following manner; any variation in the air gap (distance between shaft and electromagnets placed on the stator), detected by the position sensor, is compared as a voltage signal to the reference signal defining the nominal position of the rotor. The difference in the two signals (error signal) is treated in the PID (Proportional, Integral, Derivative) analog computer (to yield bearing stiffness and damping characteristics) which delivers an output voltage(s) to the power amplifier. The power amplifier transforms the output voltage(s) into a current which is in turn transformed into a correcting force when introduced to the electromagnet. For controlling one degree of freedom, two electromagnets are required since they operate on attraction only. For a complete rotor where five degrees of freedom must be controlled, ten electromagnets are required.

The electronic control loop is scheduled to maintain, with the highest precision possible, the rotor in its nominal position and return it rapidly through a damped movement to its nominal position after displacement by any disturbing forces.

The magnetic bearing control rack provides continuous information on the machine's operating conditions without the need for additional equipment:

1. Speed of the rotor (via magnetic pickup).
2. Bearing load and direction of the load (via current supply to bearing magnets).
3. Magnetic bearing stator winding temperatures (via RTD's).
4. Shaft displacement and vibration (via eddy current probes).
5. Eccentricity and unbalance in the rotor (via eddy current probes).
6. Capability of on site balancing of the machine at full operating speed (via eddy current probes), magnetic pickup (key phasor), and oscilloscope.
7. Hour meter (for magnetic bearing control system).

The magnetic bearing control rack provides the following trip functions:

1. Excess rotor shifting which referes to a shaft movement at any frequency.

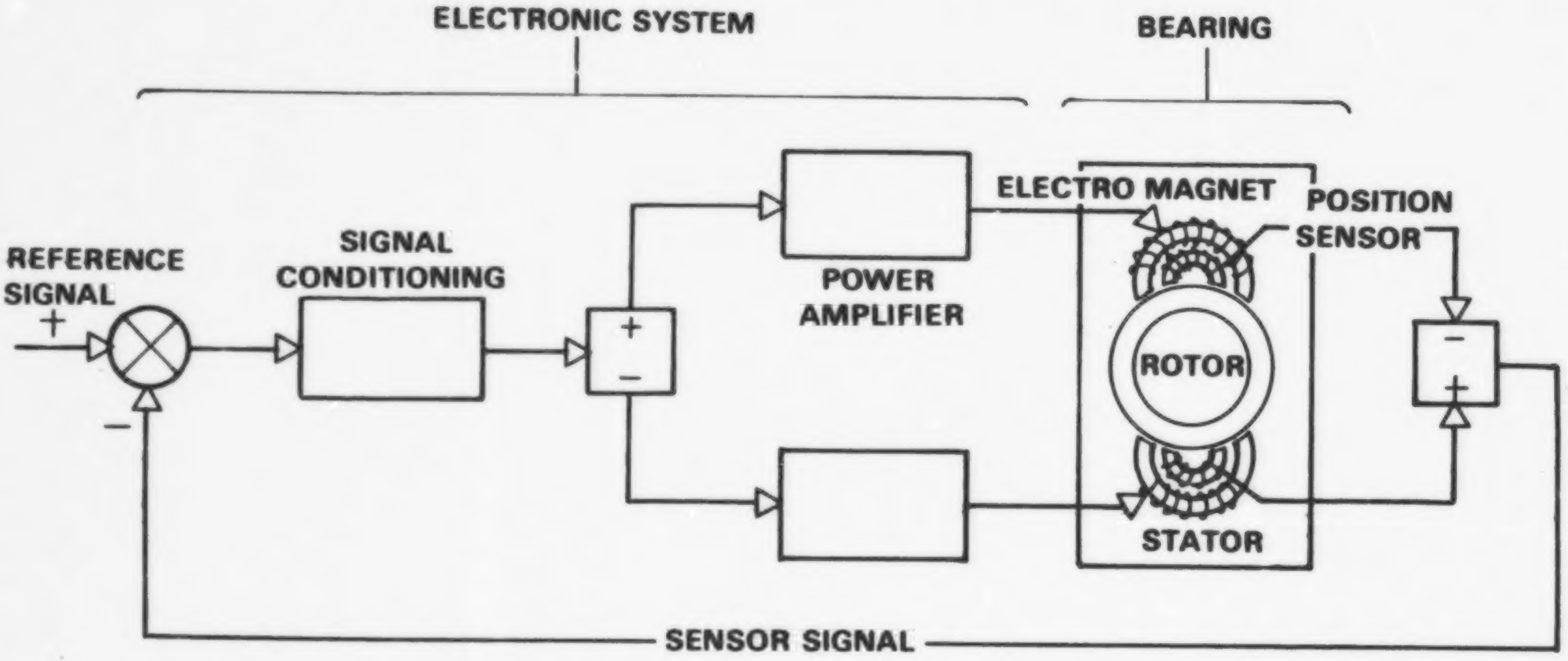

ELECTRONIC SYSTEM
BEARING
REFERENCE SIGNAL
+
−
SIGNAL CONDITIONING
+
−
POWER AMPLIFIER
ELECTRO MAGNET
POSITION SENSOR
ROTOR
STATOR
−
+
SENSOR SIGNAL

Figure 6

CONTROL LOOP
SCHEMATIC DIAGRAM

2. Excess rotor imbalance, which refers to shaft movement at rotation frequency only.

3. Excess bearing load.

4. Speed sensor failure.

The four functions listed above will cause the unit to trip off line, but the compressor will come to a levitated stop. The functions listed below will cause the compressor shaft, on shutdown, to de-levitate when the compressor speed drops below 780 RPM. This is done to protect the electronics. The rotor will slow roll from 780 RPM down to 0 RPM on the auxiliary ball bearings without damage to the machine.

1. Excess of electronic temperature.

2. Insufficient cabinet cooling water pressure.

The electronic control system is housed in the magnetic bearing control panel whose hardware includes the printed circuit boards for analysis of the shaft position sensor signals, power amplifiers and line voltage conditioning equipment. (See Figure 7)

The voltage conditioning equipment consisting of transformers, rectifiers, battery charger circuits, and batteries to convert the incoming AC line voltage into the DC voltages required by the magnetic bearing control system.

In addition, because of the high heat generated by the components in the magnetic bearing control panel, a remotely located chiller complements this control package.

5.0 THE INSTALLATION

Retrofitting of the TCPL natural gas compressor, in order to accommodate the active magnetic bearing and dry gas seals, required that the compressor be shipped to the manufacturer's plant for machining.

The casing was machined to accommodate a new, longer impeller shaft. As the pipeline conditions and the compressor performance requirements remained unchanged, the original impellers were re-fitted to the new shaft. The longer shaft accommodated the inboard and outboard dry gas seal cartridges and the magnetic bearing ferromagnetic sleeves. The shaded areas on Figure 8 depict those components that make up the dry seal and the magnetic bearing assemblies. In addition to the installation of the two magnetic radial bearings, the magnetic thrust bearing, the two auxiliary ball bearings, two dry gas seal cartridges, and two labyrinth seals, the manufacturer supplied a modified coupling and coupling guard to compensate for the longer overall shaft length.

One of the agreements in the original manufacturer's proposal to convert the compressor to magnetic bearings and dry gas seals, was the capability to revert back to the original oil film seals and tilt pad journal bearing design. This necessitated the provision of a new drive end cartridge design.

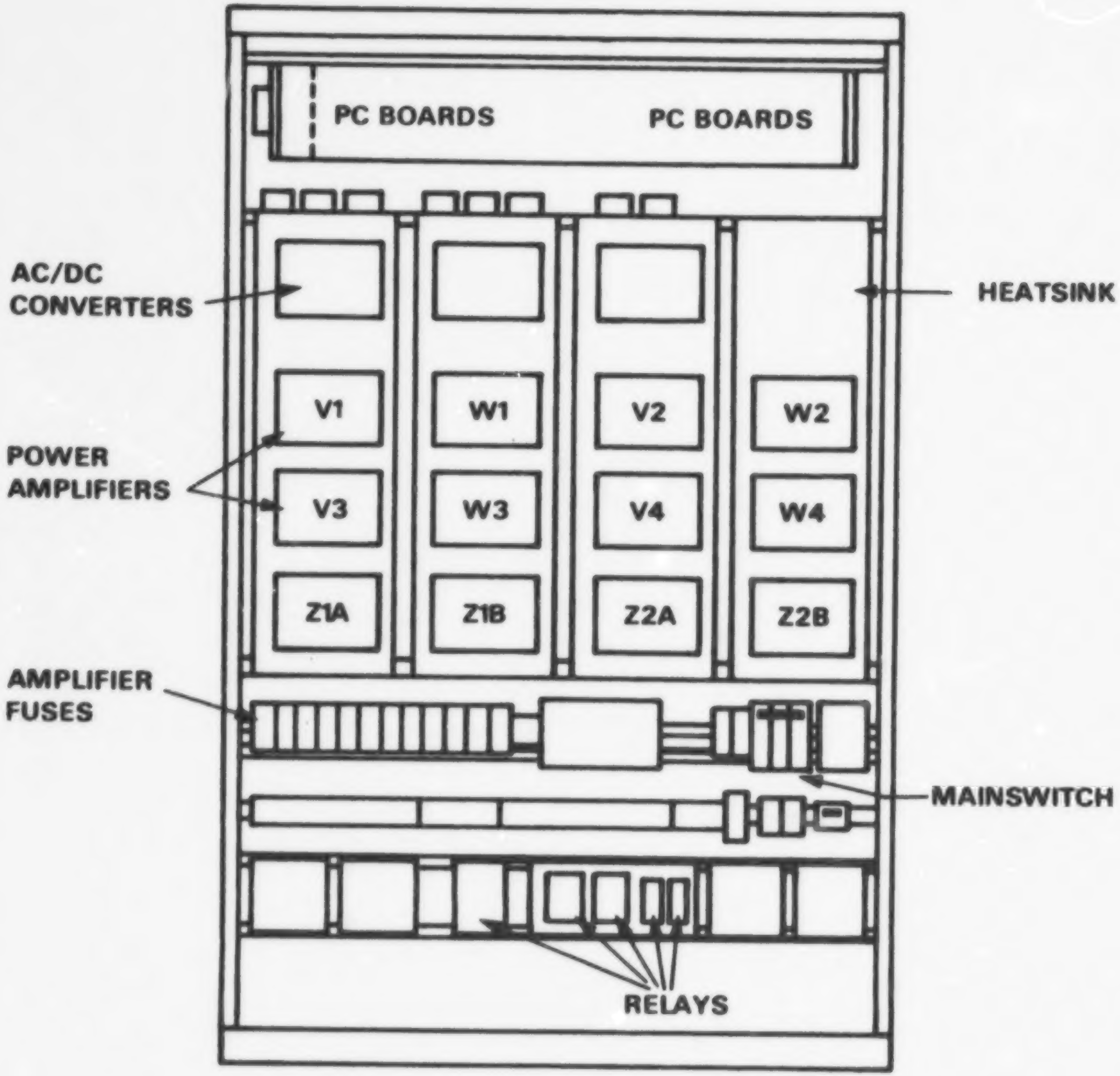

PC BOARDS
PC BOARDS
AC/DC CONVERTERS
HEATSINK
V1
W1
V2
W2
POWER AMPLIFIERS
V3
W3
V4
W4
Z1A
Z1B
Z2A
Z2B
AMPLIFIER FUSES
MAINSWITCH
RELAYS

Figure 7
MAGNETIC BEARING CONTROL PANEL

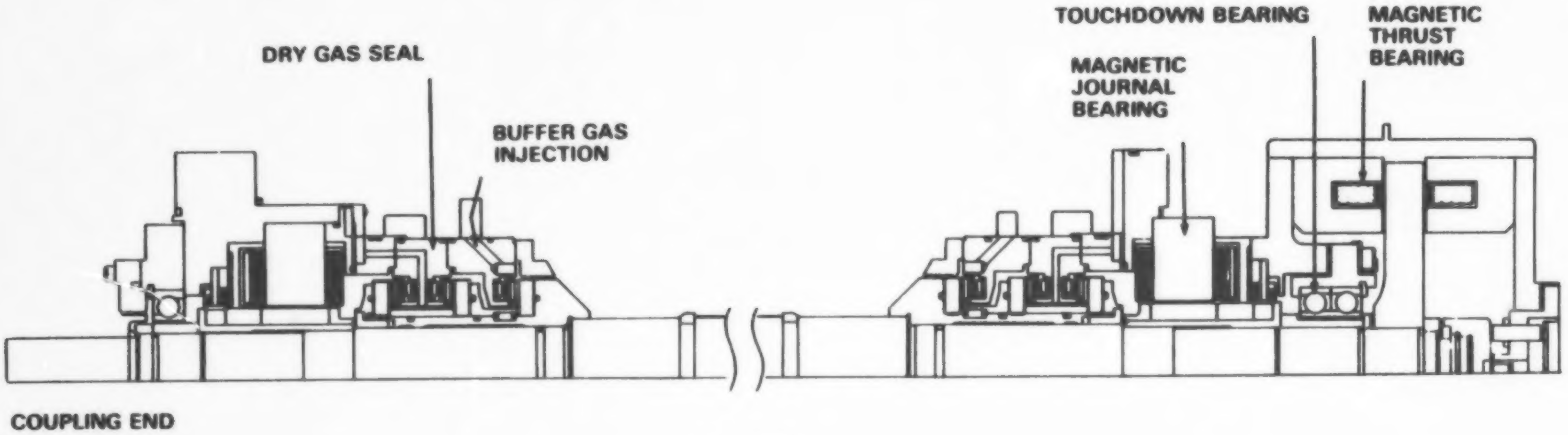

DRY GAS SEAL
BUFFER GAS INJECTION
TOUCHDOWN BEARING
MAGNETIC JOURNAL BEARING
MAGNETIC THRUST BEARING
COUPLING END

Figure 8

GAS COMPRESSOR SEAL AND BEARING ARRANGEMENT

The control package comprises a magnetic bearing control panel complete with a remote chiller package and a remote monitoring panel. This panel includes an annunciator and shutdown system to interface with the magnetic bearing control panel and the gas seal system and switch panel to shut down the gas turbine drive on a trip condition, vibration/RTD monitors to monitor position sensors and RTD's in the magnetic bearings, an oscilloscope for troubleshooting and examining shaft orbits, current readout of the individual axes and an hourmeter to record the hours of shaft levitation.

6.0 TUNING

The compressor manufacturer, in conjunction with the magnetic bearing designers, carried out an extensive tuning program on the magnetic bearings in their plant prior to offering the bearings for use in the field. The term tuning is used to describe the procedure for setting the desired bearing characteristics of a magnetic bearing.

The procedure for tuning the magnetic bearings was carried out as follows: The Magnetic Bearing Control Panel was powered-up to determine whether the correct powers and voltages were being generated. An oscillator capable of generating a 17 kHz signal was utilized for the excitation and amplifier synchronizing frequency. The sensors incorporated in the bearings were checked to ensure that the signal frequency was in phase with the 17 kHz operating frequency.

After the shaft was set manually at its mechanical or geometric centre in the auxiliary bearings, the control system was tuned so that it was initialized for this location. This was performed by changing the phase and/or amplitude of the sensor signals such that everything was zeroed.

The radial magnetic bearing characteristics were then established; a random signal generator (0 - 20,000 Hz) was used to excite the shaft by sending its signals through the panel and causing the magnetic bearings to rock the rotor assembly. This technique determined the 1st, 2nd, and 3rd, etc. criticals. Once the criticals were established, the proportional action (electrical gain which simulates bearing stiffness) and the derivative action (electrical phase lead which simulates damping) were adjusted to minimize the excitation amplitudes at these criticals. Iterative steps were required to obtain the best possible combination to minimize the amplitude at these criticals. All of the adjustments were made to the sensor control cards and the proportional integral derivative (PID) control loop cards.

The thrust bearing was tuned in the same manner as the radial bearings. In addition, the manufacturer carried out a load test on the auxiliary thrust end bearing to determine if it could handle the load and secondly to record the axial shaft displacement.

A final check was carried out following the installation of both the inboard and outboard dry gas seal cartridges.

7.0 OPERATING PROCEDURES

7.1 Dry Gas Seals

The start-up, operating and shutdown procedures for gas film seal systems are similar to those of a conventional seal oil system but require significantly less work and adjustment due to fewer components and less complexity.

The start-up sequence is permitted providing the necessary supply of seal injection gas and buffer gas is available at the required pressure level. Otherwise either of the Compressor Low Seal Gas Differential Pressure Alarms (Thrust End or Coupled End) would annunciate indicating a low differential pressure between the process gas pressure and the seal injection gas pressure.

The major operating difference is in monitoring the seal system. The gas leakage through the seal vent is monitored in the dry seal as opposed to the seal oil differential pressure in the seal oil system. Leakage across the seal would be higher during start-up than at normal operating conditions. At normal conditions, the seal will stabilize and operate at design conditions.

Shutdowns are permitted at any time with no special considerations required for relieving the sealing pressure during or after the shutdown. Essentially zero leakage occurs during a pressurized shutdown due to the combined action of the spring and the pneumatic closing forces.

7.2 Magnetic Bearings

Prior to supplying power to the magnetic bearings and the rotating machinery, the rotor is supported by the two auxiliary, dry lubricated, ball bearings at each end. The clearance between the rotor and the inner race of ball bearing was designed to prevent rotor contact with the magnetic bearing pole pieces while the rotor was at rest or during an emergency shutdown. Once power is applied to the electronic controls, the rotor is levitated by the electromagnets and the gas generator/power turbine can then be started. The sensors and control system will regulate the strength and direction of the magnetic fields to maintain the nominal rotor position by continually adjusting to the changing forces on the rotor.

To shut down the bearing system, the above procedure is simply reversed; the rotation is stopped, however the power may or may not be removed from the magnetic bearing control panel so as to leave the rotor levitated. If the power is removed, the rotor will return to rest on the auxiliary ball bearings.

8.0 PERFORMANCE

8.1 Control Loop

In the period March 1, 1986 to March 1, 1987, the TCPL compressor package experienced a total of 88 shutdowns, 47 were normal shutdowns, 25 were incomplete starts not associated with the magnetic bearings/dry gas seal conversion and only the remaining 16 shutdowns were power turbine/compressor related. Of the 16 power turbine/compressor related shutdowns, five were attributed to Compressor Low Seal Air Pressure, five to Excessive Rotor

Imbalance, one to Excessive Rotor Shifting and the remaining five to Excessive Bearing Load.

Testing of the PC Boards in the bearing control panel revealed that the Excessive Rotor Imbalance trip setting was mistakingly set lower than the Excessive Rotor Shift trip setting. The Automatic Balancing feature for this unit could not be implemented because of the compressor shaft design, coupling design and the balancing of the stacked components. Without Automatic Balancing, the Excessive Rotor Imbalance monitoring feature was deemed redundant and has since been jumpered out of service.

The shutdown due to an Excessive Rotor Shift was attributed to a faulty card in the magnetic bearing control panel. The repair report for the faulty control card concluded that the problem was simply a trim pot requiring an adjustment.

Those shutdowns triggered by an Excessive Bearing Load are suspected to be due to the high current spikes experienced in the V_2 - electrical axis of the coupling end radial magnetic bearing. The V_2 current readings are higher than the other current axis readings and have been recorded as high as 32 Amps during start-up, settling down to 28 Amps when the unit is on-line and spiking to the bearing saturation limit of 50 Amps. There is no long term detrimental effect anticipated,however, these high currents continue to indicate that the power train suffers from misalignment. An investigation is presently under way to determine the cause and degree of the misalignment.

8.2 Dry Gas Seals

The dry gas seals are now operating satisfactorily. During commissioning, the first face seal of both the Thrust End and Coupling End Seal cartridges, intermittently, were failing to seal properly and as such the intermediate seal gas pressure read the same as the booster casing pressure. The second face was sealing properly, however, because the seal gas vent registered an almost undetectable flow. Disassembly of the seal cartridge revealed small deposits of a black residue. It was found that the seals required a slight running-in period before both seal faces would seal properly, particularly during start-up.

Pipeline conditions at this site often require the compressor to operate under either a high head, low flow condition or a low head, high flow condition. The latter conditions, being the more prevalent, would inevitably raise the Compressor (Coupled End/Thrust End) Low Seal Gas Differential Pressure Alarms. These alarms have since been modified to alarm locally, thus reducing the number of outcalls to this site by personnel from the TCPL parent station. There is still, however, a general concern for the possibility of seal contamination during these conditions. Unless a separate source of seal gas is available, the situation is unavoidable.

In the event of operating in a high head, low flow condition, it may be inevitable that the Seal Gas Filter High Differential Pressure alarm will be activated. In order to avoid this situation, the alarm setting, warranting a filter changeout, can simply be raised, on the pre-caution that the risk of filters going without proper maintenance would not be increased.

9.0 ECONOMIC JUSTIFICATION

The technology of active magnetic bearings and dry gas seals is soon to become the standard in the turbomachinery industry. It was prudent that TCPL gain the experience and confidence with this new type of equipment, with the added capability of switching back to the conventional bearings and seals. In future, a new installation of a turbocompressor plant would have no such capability, as the maximum savings could only be realized if the package is bought without a conventional lube and seal oil console. The following qualitative scenario outlines the possible cost saving areas presently under review.

9.1 Dry Gas Seals

The original seal oil system comprised such auxiliary equipment as the main oil console, pumps, overhead seal oil tank, drainers, filters, degassing tank, lube oil cooler and a variety of pressure controllers, gauges and valves to compliment the oil film seals. The conversion of the subject compressor has resulted in the removal of a major portion of this equipment, therefore, TCPL is expecting to see substantial cost savings in the areas of both scheduled and unscheduled maintenance, less call outs from the parent station and less related downtime.

There are also substantial cost savings that can be realized from improved efficiency. The efficiency is increased by eliminating the parasitic losses associated with the main gear driven lube and seal oil pumps, auxiliary seal oil pump and the emergency seal oil pump. The parasitic losses due to the lube oil cooler are still present, however, as lube oil is still required by the power turbine bearings.

The installation of dry gas seals eliminates the annual cost of oil loss due to leaking seals. An increase in pipeline efficiency would not be noticed immediately after a dry seal installation unless the pipeline is cleaned and there was no further oil contamination.

Lower costs can also be expected due to the elimination of frictional horsepower in the seals. Unfortunately, any change in the compressor performance due to the new seal system was undetectable under the actual conditions of a recent Performance Test.

In the original seal oil system, the gas entrapped at the oil seal was removed by the degassing tank and vented to atmosphere. The quantity of gas being lost in this manner is substantially greater than the gas vented from the dry seals. The reduced gas losses, however small, will contribute to the reduced overall operating costs of this unit.

New installations will experience even greater cost savings due to the reduction of the total package weight and the reduction of the overall compressor building size.

9.2 Active Magnetic Bearings

When comparing the cost of the conventional tilt pad bearing system to the magnetic bearings, the latter will generally cost slightly more.

As with the dry seal system, significant savings in the operating and maintenance costs are also expected with the magnetic bearing system.

Magnetic bearings eliminate mechanical wear, therefore, allowing unlimited bearing and shaft life and thus reducing the overall maintenance costs. The parasitic horsepower losses due to the conventional bearing design have been virtually eliminated.

To fully evaluate the economic justification of retrofitting the subject compressor from the conventional tilt pad journal bearing design to an active magnetic bearing design, both the costs and savings of the oil seal system versus the dry gas seal system must be included.

10.0 CONCLUSIONS

As of April 1, 1987 the magnetic bearing control panel has maintained the compressor shaft in its levitated state for a total of 4936 hours.

During the 2108 hours of accumulated running time the following observations were made:

1. Field start-up and operation was simplified due to a minimum number of valves, controllers, vessels and electrical equipment which must be checked, calibrated and maintained.

2. The dry gas seals are operating satisfactorily.

3. The rotor behaved in a stable manner at all times when accelerating/decelerating through its first three critical speeds.

4. The rotor behaved in a stable manner while undergoing surge cycles at maximum discharge pressure.

5. Rotor operation was stable while switching from electrical power to battery power and vice versa.

6. The monitoring system of the magnetic bearing was used as a diagnostic tool for establishing and monitoring the operating parameters of the equipment. It is possible to determine the rotor imbalance during operation and rebalance the shaft's movements at operating speeds.

7. Improved performance through reduced parasitic losses within the seal itself and that associated with auxiliary equipment.

8. No oil contamination, due to the elimination of the process gas coming in contact with the lubricating oil system.

9. Any change in the compressor performance due to the magnetic bearings would be very small (around 1%), and this small change was undetectable under the actual test conditions of a recent Performance Test.

10. Minimum downtime was experienced due to the malfunctions in the active magnetic bearing system.

11. Expect lower costs due to the elimination of frictional horsepower in the seals and auxiliary equipment system.

12. Maintenance costs are significantly reduced by the partial elimination of the oil system and its inherent problems.

13. For future installations where working space is always at a premium, the reduction in space required for the lube oil console, oil cooler and other associated equipment would be significant.

The active magnetic bearings and dry gas seal conversion of the TCPL compressor package has been a mutually satisfying endeavour by all parties involved.

Although the economics of carrying out further such retrofits on other units has not been reviewed at this time, the experience TCPL has gained with this unit leaves no doubt that new larger, future units could be ordered complete with active magnetic bearings and a dry gas seal system for both the power turbine and the gas compressor in lieu of the conventional lube and seal oil system and the associated oil film seals and tilt pad journal bearings.

REFERENCES

1. Cataford, G.F., and Lancee, R.P.,
"Oil Free Compression on a Natural Gas Pipeline", ASME Paper 86-GT-293, Presented at Gas Turbine Conference and Exhibit, Dusseldorf, West Germany, June 1986

2. Hustak, J., Kirk, R.J., and Schoeneck, K.A.,
"Active Magnetic Bearings for Optimum Turbomachinery Design", Presented at Bentley Nevada Instability Symposium and the Romac Conference, San Antonio, Texas, June 1985.

3. Schoeneck, K.A., and Hustak, J.F.,
"Comparison of Analytical and Field Experience for a Centrifugal Compressor Using Active Magnetic Bearings", Presented at Third European Congress on Fluid Machinery for the Oil, Petrochemical and Related Industries, The Hague, Holland, May 18, 1987.

4. Schoeneck, K.A.,
"The Application of Gas Seals and Magnetic Bearings to Centrifugal Compressor", Presented at Pacific Coast Gas Association Transmission Conference, Spokane, Washington, April 18, 1985.

PRECISE GAS TURBINE FUEL CONTROL

Gary E. Smith
Systems Engineer
En-tronics Controls
North Sandusky Street
Mt. Vernon, Ohio 43050

SUMMARY

Recent advances in electronic circuitry and development language improvements have provided for more powerful processing capabilities with a user friendly environment. Application of this new technology to a digital fuel control will provide excellent reliability, cost effectiveness, and operational features. The digital design provides programmability thus allowing the system to be easily tailored to specific needs.

INTRODUCTION

En-tronic Controls is a division of Cooper Industries dedicated to control system design, manufacture, and service. Controls are primarily supplied for gas turbines, reciprocating engines, and associated driven equipment. Performance optimization, protection monitoring, operational reliability, and cost effectiveness are major objectives in the control system design.

The electronics technology has improved tremendously in recent years with large scale integration and low power consumption CMOS (Complimentary Metal Oxide Semiconductor). This technology has improved hardware design by reducing component count, increasing reliability, lowering cost, improving processing speed, and performing more complex tasks within a single integrated circuit. The integrated circuitry improvements further provided better development languages with which user friendly application software could be developed. These new advances in technology were utilized as the basis for the En-tronics microprocessor based Flexitrend® control system.

The Flexitrend® is modular consisting of plug in printed circuit cards. Each system will accept three parallel processors sharing a common input and output interface bus, refer to Figure 1, Flexitrend® Processing System. There is an array of analog and digital interfacing cards providing the input and output information. The operating system provides a friendly environment to write application software. A PLC (Programmable Logic Compiler) provides the sequencing which is analogous to relay ladder diagramming. Compiled math modules will perform transcendental math and logic functions which are easily written. There are modules to manage curves, PID controllers, and other specialty functions.

The processor system was first used for unit sequencing control service after it favorably assured endurance against noise and heat. The operational reliability and programmability provided a powerful system for logic and mathematical needs. A single processor provided the unit performance calculations, displayed operating curves, and the unit sequencing. Additional program requirements could be handled by other processors within the same inclusive package.

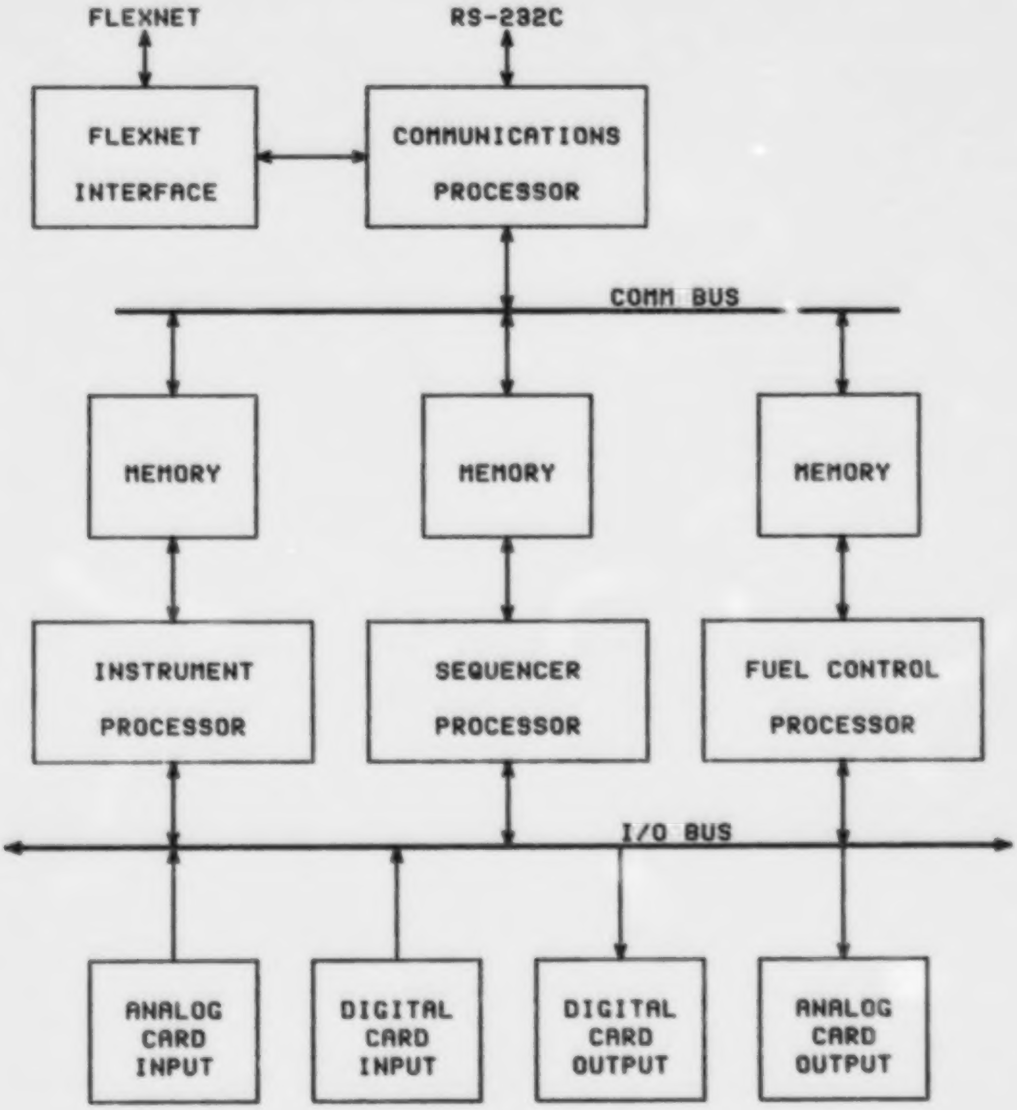

FLEXNET
RS-232C
FLEXNET INTERFACE
COMMUNICATIONS PROCESSOR
COMM BUS
MEMORY
MEMORY
MEMORY
INSTRUMENT PROCESSOR
SEQUENCER PROCESSOR
FUEL CONTROL PROCESSOR
I/O BUS
ANALOG CARD INPUT
DIGITAL CARD INPUT
DIGITAL CARD OUTPUT
ANALOG CARD OUTPUT

Figure 1 - Flexitrend® Processor System

The processing system with some additional software development provides a precise fuel control. The digital fuel control eliminates a number of problems associated with analog fuel controls, such as the rigorous calibration procedure with its resulting time delays. The digital control will be secured to protect the gas turbine operating limits and provide greater unit longevity. Diagnostics are incorporated easily within the digital control to quickly identify problems. The diagnostics will improve unit availability by identifying fuel control problems. In multiple unit application, fuel scheduling characteristics can be precisely duplicated on each gas turbine.

The turbine control system of the future will use a single integrated processing package. Let us look at how this will apply to a gas transmission turbine control unit; the PLC (sequencing), fuel control, surge control, temperature monitoring, vibration monitoring, and other necessary control functions would be handled by a single digital control system. This single system concept provides many advantages. A single communications network will report the unit data to a central processing system without setting up numerous individual communication networks. System printed circuit cards would be interchangeable throughout the package. Fewer components from a single supplier would realize greater reliability, lower costs, reduced engineering, and service time.

OBJECTIVES

In the design and development of the digital fuel control system, it was recognized that certain essential features must be incorporated. The commissioning procedure must be uncomplicated and consequently be accomplished within a short time frame. Checking and testing requirements must be held to a minimum. The fuel control must ensure better gas turbine longevity by maintaining operating limits, precisely holding schedules and avoiding excessive temperature transients. It must provide communications interfacing thus allowing data networking. It must incorporate diagnostics that will allow quick resolution to problems within the processing system or associated fuel delivery equipment. With these objectives established, the fuel control design began.

DESIGN

The specific fuel control design activities will be described in the appropriate sections. The processing system required additional features and some of the features were enhanced. A hydraulic actuator was developed to function in conjunction with the digital control. A fuel metering valve was obtained that would adapt to the actuator. The program was developed to emulate an analog control with digital processing advantages.

PROGRAMMABLE MODULE

The Flexitrend® fuel control is provided in two package configurations; integrated within the turbine unit control, or a Programmable Module. When the fuel control is provided in a integrated package, it has a dedicated processor and shares input and output interface with other processors. The rack mounted Programmable Module was introduced so that the fuel control could be installed in a self contained module, refer to Figure 2, Fuel Control Programmable Module.

The processor architecture contains an array of modular programs for system operation and specific tasks. For example, there are tasks to interface digital and analog data, curves, and PID controllers. The fuel control required new tasks; a setpoint driver module, PID controller with low signal selection module, and curve modules. These newer modules had to be developed so that execution time was optimized.

The processor hardware design electrically isolates digital and analog data inputs and outputs for system protection. The analog conditioning circuitry is adjustable for exacting calibration. This analog circuitry maintains long term calibration through the use of stable power supplies and newer analog technology. The fuel control required an actuator driver and LVDT (Linear Variable Differential Transformer) position conditioning hardware.

The standard communications link is RS-232C (serial) but the high speed networking communications (Flexnet) could be provided. The Flexnet is an Entronics local-area-networking system allowing "master-slave" and "peer-to-peer" data exchange at 2.5 million bits per second.

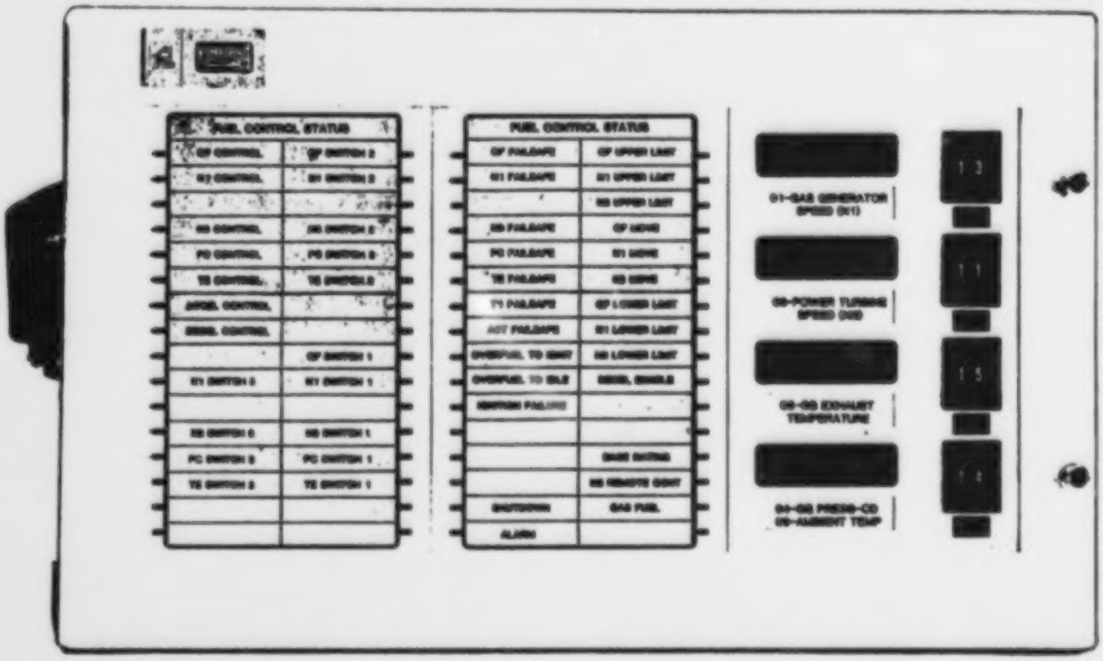

FUEL CONTROL STATUS
FUEL CONTROL STATUS
01-GAS GENERATOR SPEED (N1)
02-POWER TURBINE SPEED (N2)
03-GG EXHAUST TEMPERATURE
04-GG PRESS-CD
05-AMBIENT TEMP
SHUTDOWN
ALARM
GAS FUEL

Figure 2 - Fuel Control Programmable Module

The processor hardware diagnostics provides LED indication and failure detection. This diagnostics scheme in conjunction with a simple instructional reference will pin point most problems very quickly. The software diagnostics relies on a programming terminal, IBM Personnel Computer or compatible. The programming terminal provides the ability to view memory values. The fuel control provides display data maps that will be observed with the programming terminal for most data. The fuel control also provides digital displays to read the significant parameter channels.

ACTUATOR

The hydraulic actuator was sourced through a recognized manufacturer with gas turbine fueling experience. The actuator will close the rigidly coupled valve in under a 100 milliseconds. Operating temperature limit is 300 degrees fahrenheit. Operation is permitted in a vibratory environment of up to 10 G's to 1000 Hz. The cartridge style transformer LVDT is easily replaceable. Actuator force magnitude is 4 times greater than required by the valve. The integrating style actuator contains no resonance producing components, such as springs, within 10 times the operating frequency. The digital fuel controller and actuator have a 70 degree open loop phase margin and 5.25 Hz. close loop frequency response.

VALVE DESIGN

The fuel metering valves were sourced from a recognized valve manufacturer, refer to Figure 3, Actuator and Valve. There is a complete range of tight shutoff balanced trim valves available. All valves have 3/4 inch stroke and common mounting for both liquids and gaseous service. There exists no linkage looseness between the valve and actuator. The valve stem is screwed into the actuator cylinder and locked.

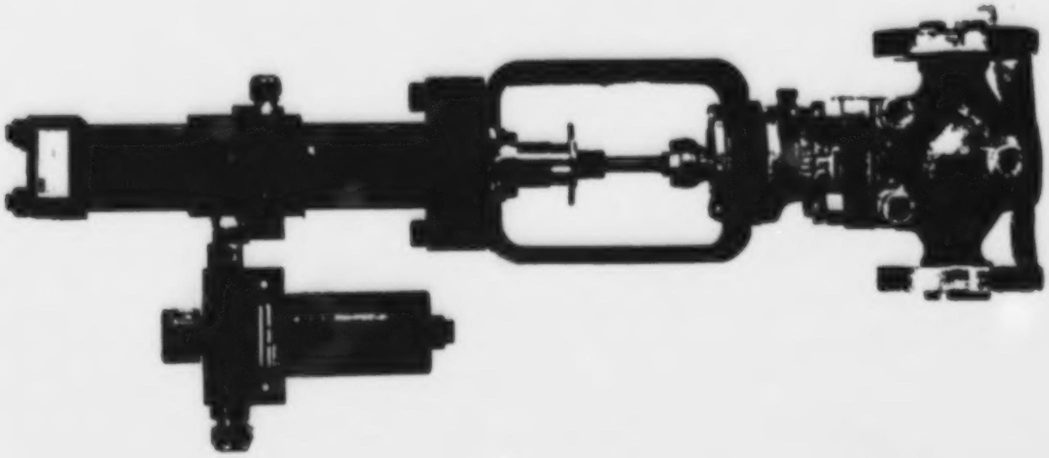


Figure 3 - Actuator and Valve

FUEL CONTROL PROGRAM

The fuel control program consists largely of "Read Only" floppy disk files for security. Once a fuel control program has been proven by extensive testing, modification should be performed by the originating organization. The gas turbine limit schedules are secured within the program. Customer site application variables are adjustable within protected ranges. The program and associated data are contained on one 5-1/4 inch floppy disk.

The fuel control operating system contains four programs; tuning, down load, dump, and communications. Tuning provides the ability to "On Line" tune the PID controllers and adjust the user constant values. Tuning includes display maps to conveniently monitor operational values. The down load program provides the means to restore the fuel control program into the battery supported processor memory. The dump program allows the processor memory to be viewed. The communications program provides selection of host and peer values. This operating system is supplied on another 5-1/4 inch floppy disk.

The fuel control program contains two sections; system configuration and time class task executable list. The system configuration defines; controller memory usage, inter-controller communications, and peer communications. The time class task execution is divided into three groups; first, second, and background. The fuel control operates the first time class every 10 milliseconds and operates the second time class every 70 milliseconds. The background operates with the remaining processor time which is typically twice a second. Each time class has a list of executable tasks which are selected from a menu. This task menu contains input, output, setpoint driver, PID controller with low signal select, curves, math modules, PLC logic modules, displays, and more. The processor enters the first time class task list and performs the tasks in order until completion, waits for the next 10 millisecond interrupt and reruns. The second and background time class task lists are performed in a similar manner except they are time sliced.

The math module performs the mathematical and logic functions. The fuel control utilizes twelve math modules, each performing specific functions. The math module is structured requiring the variables and digital mnemonic to be declared and assigned a memory location number. The mnemonic is then utilized in the equation or logic statement. Each controller has 4096 analog and 4096 digital channels which can be cross referenced through out the time class listings.

The fuel control contains acceleration and deceleration limit schedules provided by the engine manufacturer. The acceleration schedule is based on compressor delivery pressure verses fuel flow. The deceleration schedule is based on LP compressor speed verses fuel flow. The fuel flow in the past has been inferred from valve position. This approach requires exacting valve trim flow calibration and necessitates knowing the upstream supply gas pressure. The long term accuracy is not known. The Flexitrend® fuel control utilizes manifold pressure which provides a consistent relationship between similar engines providing they utilize the same burner orifice size. The manifold pressure approach utilizes two pressure transmitters; a low range for ignition and operation to idle, and a high range for operation from idle to load.

The fuel control in conjunction with the low range fuel manifold pressure transmitter and fuel manifold pressure schedule set the ignition fuel pressure. This fuel manifold pressure schedule provides a protected range with which the engine can obtain ignition. Ignition is detected by exhaust temperature rise rate and fuel flow is minimized to eliminate excessive exhaust temperatures previously associated with the fixed ignition fuel settings of analog fuel controls. This approach should eliminate ambient temperature sensitivity and corresponding failed starts.

The fuel control has two status and a digital readout display modules. Each status display contain 32 LED indicators which provide a quick report of operational and failure information. The failure indicators remain latched until reset. The digital readout module has four easily read displays for speed, temperature, pressure, actuator position, and fuel flow.

TESTING

The fuel control development program required three simulation tests to be performed. These tests were performed at En-tronics development facility, the actuator manufacturing plant, and Rolls Royce. The En-tronics test simulated a gas turbine, the actuator, and the low speed starting sequence. The actuator simulation tests checked the primary control loop. The Rolls Royce test were run with a gas turbine simulation. Each testing phase was invaluable in proving the design.

Each fuel control delivered has been component and system tested. The printed circuit cards are operationally tested and the analog interface cards are calibrated. The fuel control system testing is very important and provides the confidence to operate a gas turbine. The fuel control program demands simulation testing under varied conditions to thoroughly prove intent. There should be no major program flaws after the simulation phase.

The fuel control operational testing will confirm that the gas turbine and associated driven equipment is fueled properly for the intended application. The gas turbine loading rates and PID controller tuning is site adjustable within protected limits to account for the driven equipment characteristics. The operational testing phase provides knowledge for improved simulation and ideas where improvements will be made. The testing will be reported chronologically from the early development testing to achieving an operational fuel control.

NOVEMBER 1985

The prototype fuel control was tested on an analog simulator at En-tronics. This proved that the logic and analog data flow were correct. Excessive time was required to run some modules which forced them to be written in Assembly language. A high speed oscillographic recorder was utilized to observe response. Controller tuning was not felt to be correct but at least a starting point for future testing.

APRIL 1986

The prototype fuel control was operated on a RB211 simulation at Rolls Royce, Ansty. The RR simulator programs have been proven to closely correlate with actual test bed operation. The simulation was valid from idle to full load operation. The controller tuning was optimized which changed the tuning settings slightly from the previous Entronics settings. The simulation demonstrated that a setpoint driver tracking feature was interfering more than assisting operation and was eliminated.

JULY 1986

A Coberra 3000 Gas Turbine was operated with a water dynamometer on Mt. Vernon Rotating Test Facility. The fuel flow was intended to be read by a mass flow meter but the meter was found extremely sensitive to vibration. The meter was abandoned and the fuel control program was written to use fuel manifold pressure and differential pressure across the fuel burners. The unit then started smoothly with minimal exhaust temperature level. LP engine control was stable through out the speed range. The power turbine speed control was some what unstable at low power levels concurrent with gas generator bleed valve operation. However, power turbine speed was stable at higher power levels.

The acceleration and deceleration limiter controllers had to be incorporated into the control scheme to convert terms. A great effort was spent trying to optimize the controller settings. Since the dynamometer is loaded manually, the fastest the unit could be loaded from idle was 45 seconds. The inability to accelerate faster cast doubt on the controller settings. These settings would have to be proven on a simulator or test bed runs.

The gas turbine was operated 5 hours with the fuel control. This test confirmed the viability of the digital fuel controls. The engine operation was acceptable after several initial program inconsistences were corrected. The ability to hold speed on control was good except for the above described. The testing was oscillographically recorded for latter study.

AUGUST 1986

The fuel primary control loop was closed with the Parker Actuator. This test was conducted at the manufactures facility with the fuel control hardware and a test program. The primary controller tuning was determined and response time measured. The positioning accuracy and repeatability was found to be excellent.

SEPTEMBER 1986

Coberra 2000 fuel control simulation testing was conducted on a production unit in the Mt. Vernon En-tronics plant for delivery to a major Canadian gas transmission company. The gas turbine simulation program was written digitally within the same processor. Simulation testing demonstrated that the primary controller tuning achieved during the actuator testing was not going to be stable. The actuator testing demonstrated that a derivative action primary controller was needed. The simulation testing also demonstrated that a integral action primary fuel controller was needed with the secondary controllers.

The fuel control utilized position feedback to infer fuel flow. The Fisher valve was flow calibrated utilizing water as the media. The water testing provided a trim area that was utilized in the burner calculation to resolve fuel rate against valve position.

OPERATION

A fuel control instruction book was written to describe the installation, operation, and diagnostics. The following major points were described. Digital equipment requires immunity from electro magnetic interference and static electricity. The electrical installation requires shielding or distance separation from interference. After the fuel control installation, the shielded cabling requires checking to assure that the shields are floating. Printed circuit cards require care to avoid static discharge from touching them. The hydraulic oil supply and drain lines sizes were specified.

The fuel control was commissioned in November 1986, on a Canadian gas transmission station, within a week. This installation required additional time to assure that the control was correct prior to the unit being operated. The primary fuel control loop tuning and starting fuel sequence checks did not require the extensive testing. A data recorder and oscilloscope was utilized to monitor the actuator feedback and the starting fuel manifold pressure while starts were initiated with the ignitors disconnected. This was performed numerous times until the proper tuning had been achieved. The unit started on the first attempt and came to idle, accelerated to obtain power turbine break away, and returned to idle for power turbine warmup.

The unit has operated 4230 hours on the gas transmission service. The spare fuel control and spare actuator have operated about half the accumulated hours. The actuator and fuel control change out was found to be an expedient set up. The gas turbine starting has been repeatable with minimal exhaust temperature excursion. A lower ignition speed was employed and has found to be better. The actuator maintains position within half percent total stroke on steady state operation. The fuel control has been easily adaptable to customer requirements. The fuel flow inferred measurement is going to be converted to fuel manifold pressure to provide program uniformity.

There have been problems associated with the fuel control installed on site. Three power supplies have failed; 5 VDC, 12 VDC and +/- 15 VDC, all are powered from the station invertor and were in service when that invertor failed. These power supplies are primary switching type and the problem is being carefully reviewed.

The gas generator speed pickup in conjunction with the fuel control speed conditioning card, caused a speed upset that produces a resulting short time duration 50 degree fahrenheit disturbance when the engine operates at approximately 7280 RPM. A digital filter has been installed to eliminate the disturbance with out impairing speed responsiveness.

A fuel control start permissive value failed to operate. The processor was reset and the functioning returned to normal. Status output relays required additional dead band to eliminate relay chatter and allow customer interfacing parameters to operate more smoothly.

CONCLUSIONS

The processor based fuel control will provide advantages over the previous analog systems. The repeatability, simple commissioning procedures, and adaptability to customer needs are the major advantages. The fuel control has a favorable response from users.

Integrated control systems provide better communications, printed circuit card interchangeability, reliability, and cost effectiveness. These integrated packages will be the systems of the future.

ACKNOWLEDGEMENTS

Parker Hannifin Corporation
Fisher Controls

TECHNOLOGY TRENDS IN AIRCRAFT GAS TURBINES AND THEIR INDUSTRIAL APPLICATIONS

Sam P. Sampath
Chief, Combustion Aerodynamics
Pratt & Whitney Canada
P.O. Box 4080
Mississauga, Ontario L5A 3Z4

SUMMARY

Technology trends in small aviation gas turbine engines are presented, which have been dictated by demands for improved performance and lower operating costs. Some of the technologies will find application in future industrial gas turbines used in pipeline and energy sectors. Canadian research activities in gas turbine component fields such as compressors, turbines, combustors and materials are reviewed briefly. Application of technology advancements to new engine design is examined in relation to Pratt & Whitney Canada's new PW100 engine. Potential for industrial application of PW100 is described.

INTRODUCTION

The energy crisis of the '70's and higher fuel costs have triggered development of new series of aircraft gas turbines which are more fuel efficient, and represent improvements to reliability, durability and ownership costs. These advancements have potential applications on a wide range of fixed and rotary wing aircraft used in commercial, military and general aviation. A number of previous and current industrial gas turbines have had their origins in aircraft engines, and the current developments are also likely to find industrial application. Improvements to overall performance have been achieved through better optimization of operating cycles, the use of more rigorous design and analysis techniques for components, and application of advanced materials technology to critical areas. Developments in critical components such as compressors, turbines and combustors are presented in the context of small aircraft gas turbine engines and their potential application to industrial gas turbines is discussed.

SMALL AVIATION GAS TURBINES & THEIR INDUSTRIAL DERIVATIVES

The gas turbine engine is well accepted as an industrial power plant with characteristics equivalent or superior to other types of prime movers - advantages are in space requirements, weight, reliability, fuel tolerance, cold starting, vibrations, ease of installation and environmental impact. Several of the industrial gas turbines have been derivatives of aircraft power plants which have gone through extensive development to achieve the same characteristics. An example is the ST6 engine which is a derivative of the well established PT6 Aero engine produced by Pratt & Whitney Aircraft of Canada. The ST6 engine has been used in a wide range of applications covering marine propulsion, electrical power generation, mobile power units and traction applications from trucks to trailers (1). It was also used in a specially designed car for the Indianapolis racing circuit(2).

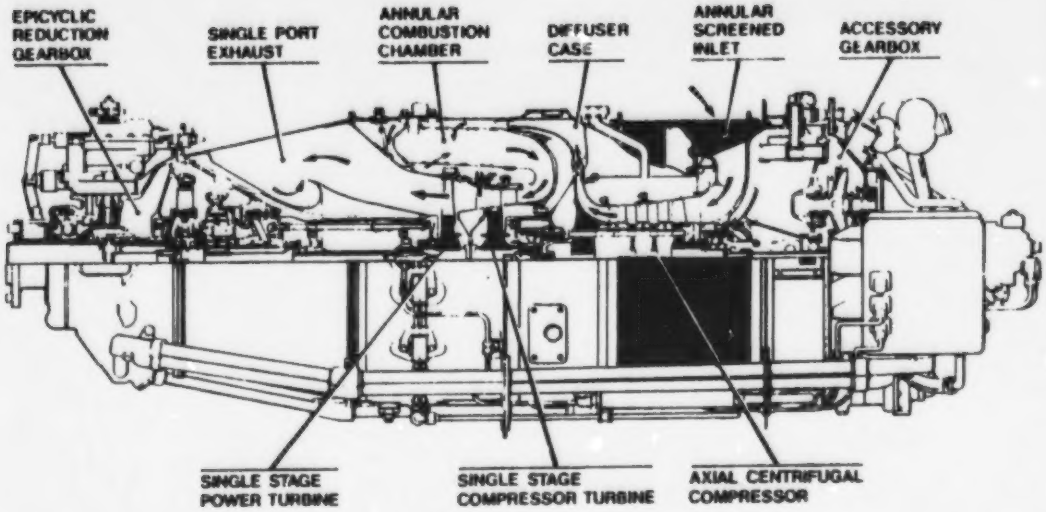

EPICYCLIC REDUCTION GEARBOX
SINGLE PORT EXHAUST
ANNULAR COMBUSTION CHAMBER
DIFFUSER CASE
ANNULAR SCREENED INLET
ACCESSORY GEARBOX
SINGLE STAGE POWER TURBINE
SINGLE STAGE COMPRESSOR TURBINE
AXIAL CENTRIFUGAL COMPRESSOR

Figure 1 - ST6 Engine Cross Section

Figure 1 shows the aerodynamic features of the ST6 power plant. The gas generator section comprises three axials followed by a centrifugal compressor, a reverse flow annular combustor and a single stage axial turbine. The power section has a single axial turbine stage on a separate shaft, single port exhaust and an epicyclic reduction gear box. The maximum output horsepower covers a range from 550 to 1065 with a 1850 HP capability as a 'Twin-Pac' version (Table 1). The performance characteristics are ideally suited to many types of industrial equipment, the free turbine allowing a wide range of output speeds at constant output power and with capability to readily accept overloads. Figures 2 and 3 show ST6 application in gas compressor units and electric generator sets.

Table 1: Power Ratings of ST6 & SPW Engine Models

MODEL	RATINGS @ 15°C			SFC @ Max. Power (15°C Sea Level)
	Maximum	Intermediate	Normal	
ST6 - 60	550	445	390	0.670
ST6 - 70	620	580	510	0.640
ST6L - 81	1065	955	840	0.580
ST6T - 76	1850	1645	1440	0.615
SPW 124-1	2810	-	2440	0.476

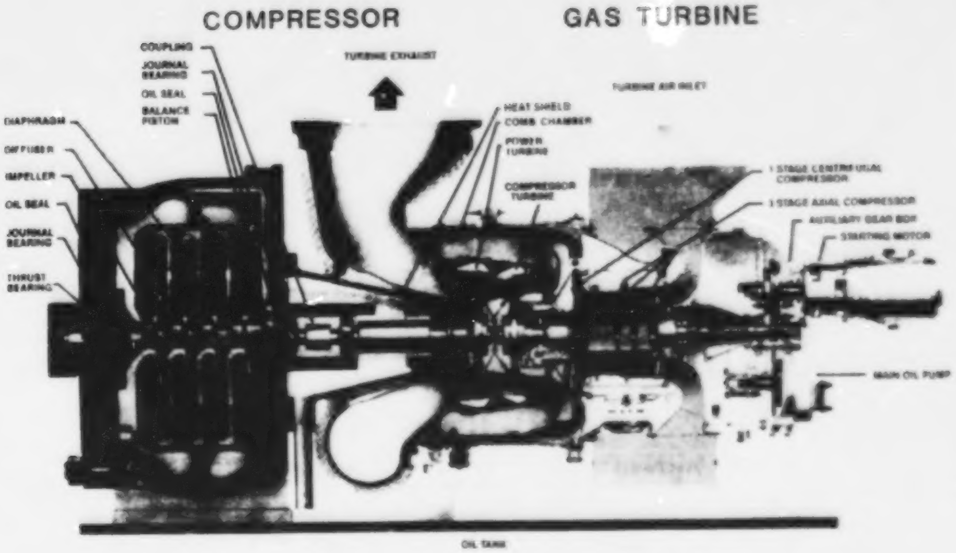

COMPRESSOR
GAS TURBINE
COUPLING
JOURNAL BEARING
OIL SEAL
BALANCE PISTON
TURBINE EXHAUST
DIAPHRAGM
DIFFUSER
IMPELLER
OIL SEAL
JOURNAL BEARING
THRUST BEARING
HEAT SHIELD
COMB CHAMBER
POWER TURBINE
COMPRESSOR TURBINE
TURBINE AIR INLET
1 STAGE CENTRIFUGAL COMPRESSOR
3 STAGE AXIAL COMPRESSOR
AUXILIARY GEAR BOX
STARTING MOTOR
MAIN OIL PUMP
OIL TANK

Figure 2 - ST6 Engine Application for Gas Compression

To meet special requirements of industrial applications, the PT6 aero engine was further developed in the following areas:

1. Corrosion Protection: A number of engine components are subjected to corrosion protection treatment (Table 2). In some components such as compressor casings, rotors etc., corrosion resistance has been enhanced by application of coatings, whereas in others improvements in materials technology have provided the same benefit.

2. Compressor contamination can result in performance loss while operating in an industrial environment. Compressor washing facilities with demineralized water and industrial detergents ensure compressor performance over long periods of time.

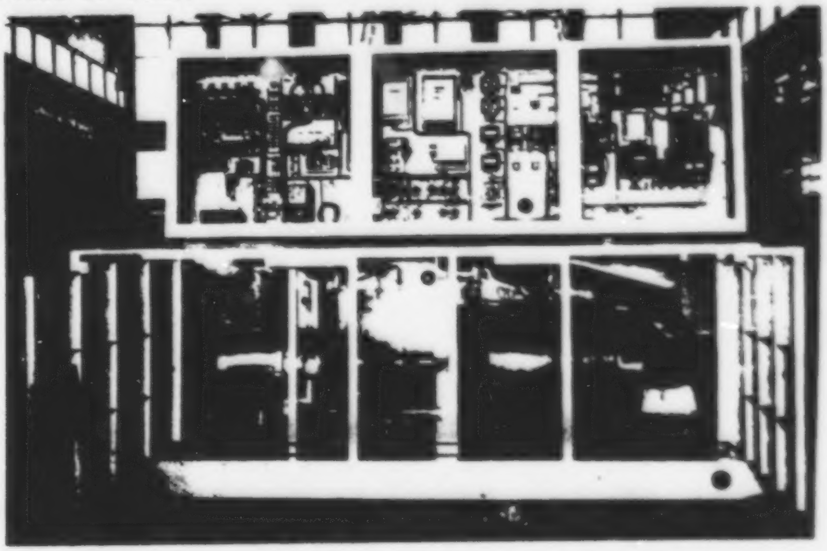


Figure 3 - ST6 Powered Electric Generator Set

3. Accomodation of high viscosity liquid fuel and gaseous fuels: Changes to the fuel manifold and fuel injection system ensure adequate performance with heavy fuels such as diesel, as well as gaseous fuels such as natural gas. Cold ignition with high viscosity fuels can be readily achieved with air injection into the fuel manifold. Both natural gas and propane can be used with reconfigured fuel injectors.

4. Erosion protection control under dusty environments is achieved using special inertial separator filters.

5. Automated controls to ensure continuous engine monitoring and long time (unmanned) operation.

Table 2: Corrosion Treatment for ST6 Industrial Turbines

Parts	Basic/Early Material	Anti-Corrosion Treatment/ Material Changes
Compressor Casing	Martensitic Steel	'Sermetal W' Aluminum Coating
Exhaust Casing	Martensitic Steel +'Sermetal W' Coating	Ni Base Alloy (IN 625/718)
Compressor Rotor	Martensitic Steel	'Sermetal W' Aluminum Coating
Compressor Impeller	Martensitic Steel	Titanium Alloy
Compressur Turbine Vanes	CO Alloy with Aluminum Coating	INCO 792
Compressor Turbine Blades	INCO 713 with Aluminide Coating	INCO 792
Castings	Magnesium	Aluminum or Titanium

Some of the applications of the ST6 are listed in Table 3 with total running time approaching 2.5 million hours. The ones that have found extensive application are mechanical drivers for gas compression and pumping, propulsion for fast boats and air cushion vehicles and auxiliary power units for ground and sea power. Descriptions of the early applications of ST6 can be obtained from Reference 1.

Table 3: Some Applications of ST6 Gas Turbines

Application	Details	Customer
Gas Compressor Units	ST6 Powered Gas Pumping	Sulzer Turbosystems International
Generating Units	ST6 Powered air transportable generator sets	"
ACV Propulsion	Shipboard ACV	Bell Aerospace Canada
Auxilliary Power Units	Electrical & Hydraulic Services	Grumman Aerospace

TRENDS IN SMALL AVIATION GAS TURBINE TECHNOLOGY

Specific Fuel Consumption (SFC), engine weight, first cost, operating cost and reliability are the primary requirements influencing developments in aircraft gas turbine engine technology. For the engine designer these requirements often result in conflicting choice of decisions about an optimum operating cycle or level of technology. For example, for small business aircraft operation, first cost is most important followed by fuel cost and maintenance cost. On the otherhand, for a large turbofan engine used in long range applications, operating cost becomes the most important factor and a high level of technology (or high first cost) may be acceptable. For helicopter applications weight is the critical factor, whereas cost of fuel may not be a large part of the operating costs. For stationary applications, reliability is the most important factor, followed by operating costs and in some cases ease of installation. Life-cycle cost, reliability and maintainability are all inter-related but also have individual influences. For example, a very complex high temperature engine may be very good on fuel consumption, but may have frequent maintenance problems reducing the time between overhaul and increasing maintenance costs. A wide range of technology improvements are therefore necessary to adopt engine designs to individual applications and achieve specific demands on engine weight, specific fuel consumption, first cost, operating cost and reliability.

Component research has resulted in technology gains which offer a range of choices to the engine designer. Trends in small gas turbine technology in the areas of compressor, turbine, combustor and materials are reviewed in the following sections.

Compressor Technology: High efficiencies in small engine compressors have been achieved using relatively few numbers of axial stages followed by a centrifugal stage (Figure 4). Whereas for pressure ratios up to 8:1 single centrifugal compressors have high efficiencies and extremely good reliability, performance of higher pressure ratio compressors have been further improved

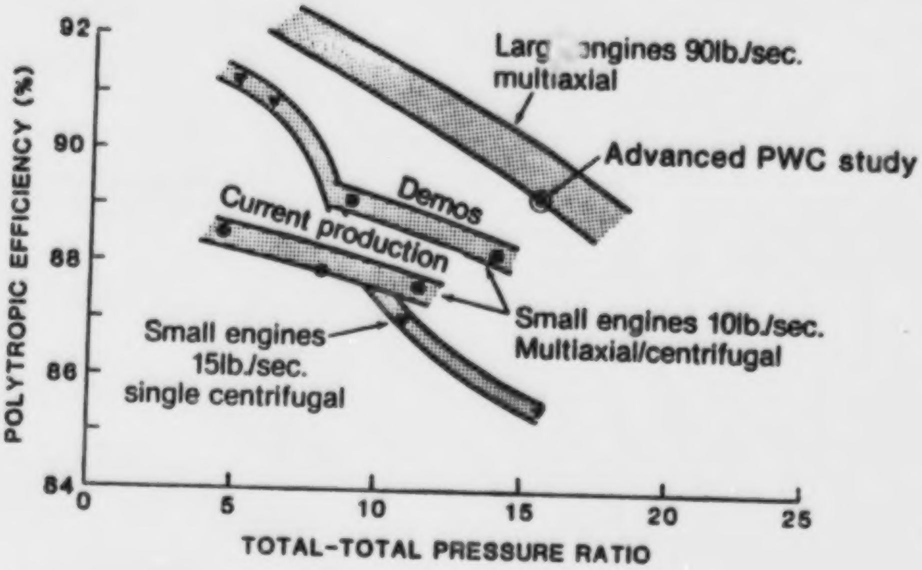
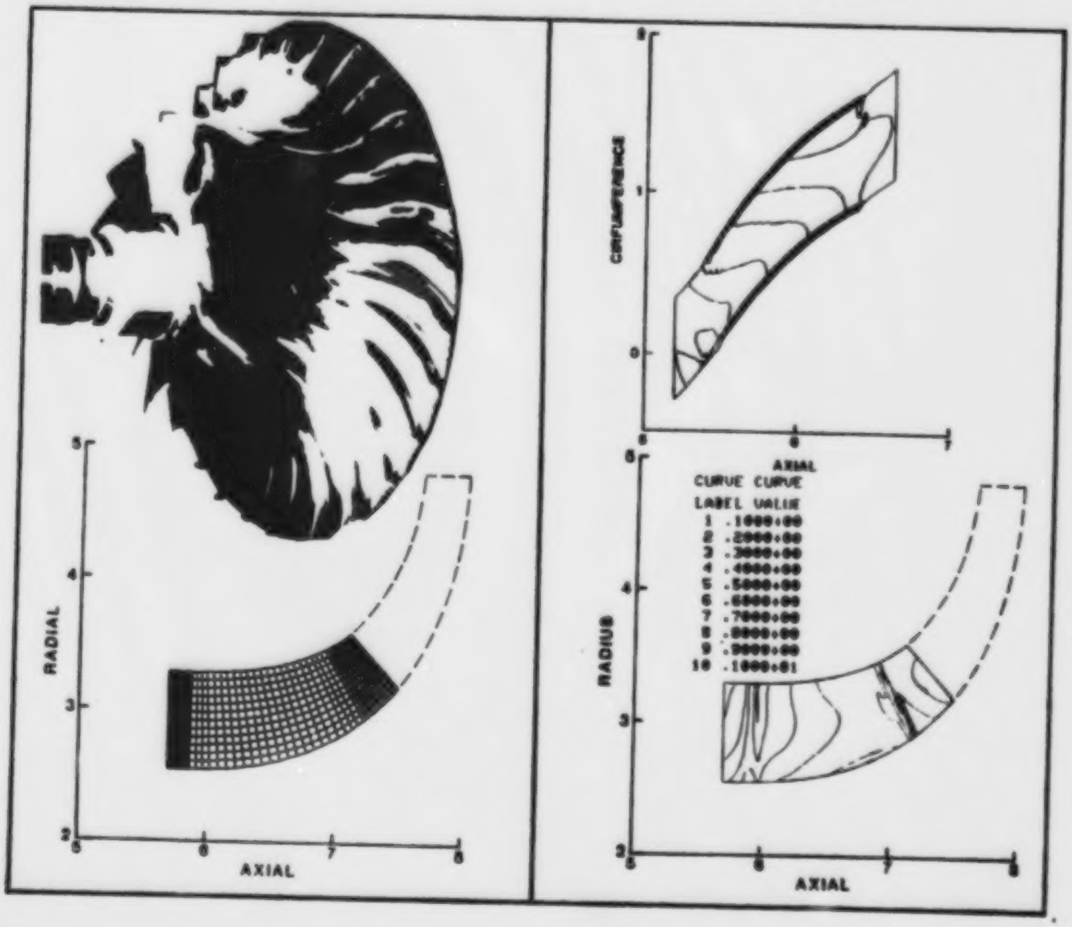


Figure 4 - Trends in Small Gas Turbine Compressor Performance



RADIAL
AXIAL
5
6
7
8
2
3
4
CIRCUMFERENCE
0
1
AXIAL
RADIUS
CURVE CURVE
LABEL VALUE
1 .1000+00
2 .2000+00
3 .3000+00
4 .4000+00
5 .5000+00
6 .6000+00
7 .7000+00
8 .8000+00
9 .9000+00
10 .1000+01
AXIAL

Figure 5 - 3-D Flow Modelling of Impellers

by optimizing multiaxial/centrifugal designs. Large engines tend to use all axial designs, but for small engines these would be inefficient due to reduced size of blading. Improvements in centrifugal compressor technology have enabled small engine compressor efficiencies to approach those achieved by all-axial compressors used on large engines (Figure 4).

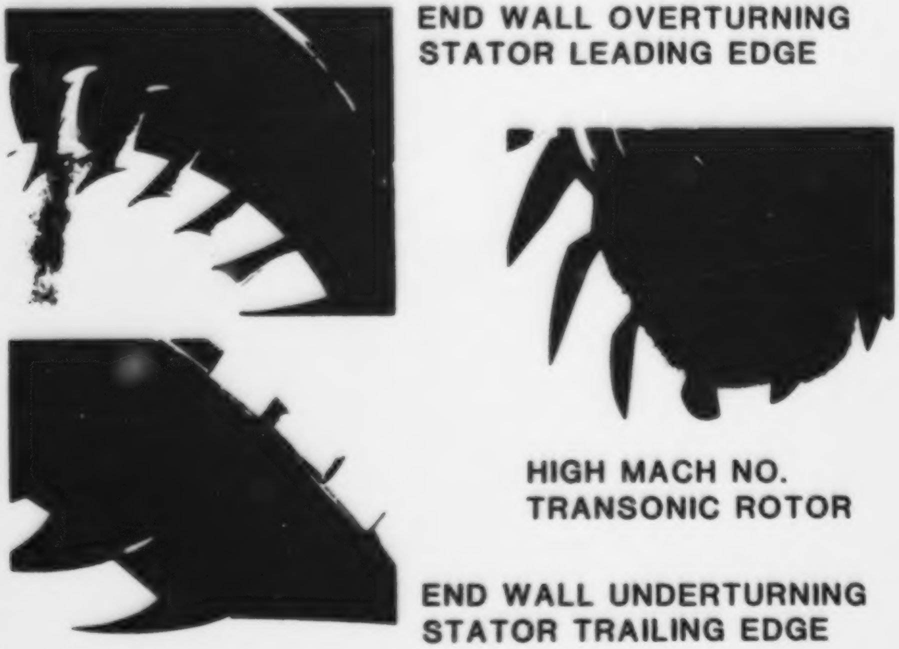

END WALL OVERTURNING
STATOR LEADING EDGE
HIGH MACH NO.
TRANSONIC ROTOR
END WALL UNDERTURNING
STATOR TRAILING EDGE

Figure 6 - Three Dimensional Designs of Axial Compressors

Full and quasi 3-D analytical methods are being increasingly used in compressor aerodynamic designs. Figure 5 shows Mach number predictions in the complex inducer section of a centrifugal compressor using 3-D modelling. Channel flows between blades can be analyzed in detail highlighting regions of loss

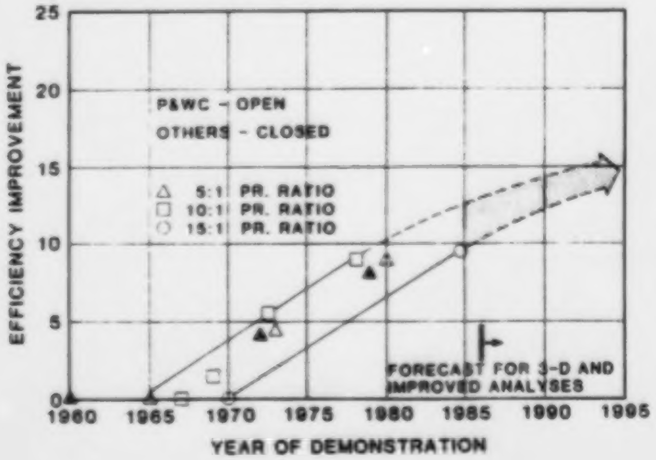


Figure 7 - Trends in Centrifugal Compressor Efficiency

due to shock/boundary layer interactions, end wall effects, flow blockage and deviation. Quasi 3-D modelling of axial compressor blades have also resulted in airfoil/boundary layer optimization for the complex blade geometry, and development of high Mach No. transonic rotors (Figure 6). Further use of 3-D viscous program is expected to give 3-4% increase in aerodynamic efficiency with no penalty in cost, weight or life (Figure 7). Mechanical design and manufacturing method improvements should result in reduced leakages and better quality of blading. One piece compressor rotors (blisks) are being machined which again will enhance structural performance of the compressor at a reduced cost.

Turbine Technology: Turbine efficiencies have also benefited from strides in Computational Fluid Dynamics (CFD). Figure 8 shows a seven point gain in turbine efficiency since 1965, at least two of which are due to development of 3-D design techniques. Full 3-D viscous modelling of airfoil flows is expected to provide further efficiency gains in the coming years. Complex blade and nozzle shapes, to control and manipulate three dimensional pressure gradients, can be designed by the use of these techniques (Figure 9). Performance of small size turbine blades is heavily penalized by tip clearances and use of minishrouds can reduce clearance losses without the structural limitations of fully shrouded blades (Figure 9). Large turbofan engines

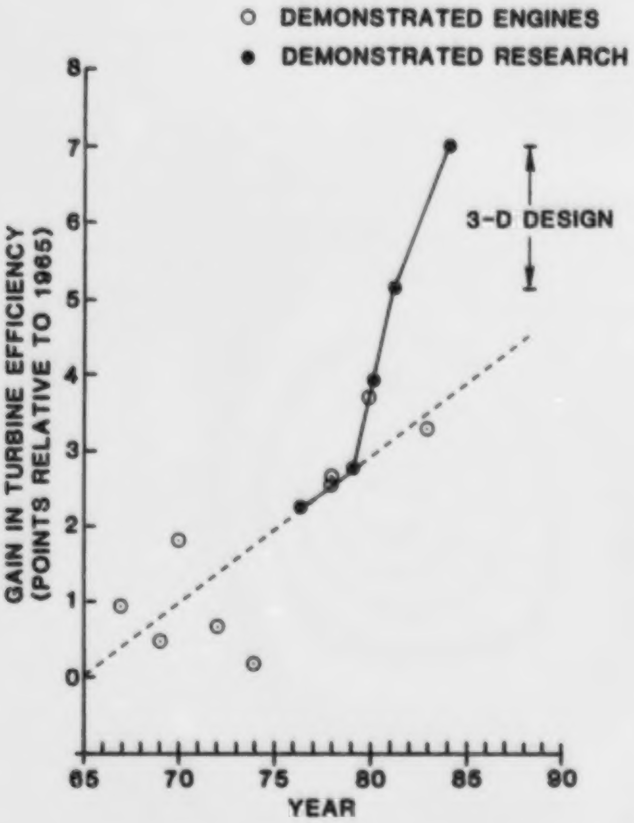


Figure 8 - Trends in Small Axial Turbine Efficiency

have some form of active tip clearance control using variable amounts of cooling air, but this generally is too expensive for small engine application. As the engine cycle demands on turbine inlet temperature increase, turbine blade cooling will get more sophisticated, using internal convection cooling passsages optimized to maximize heat transfer rates (Figure 10). New technologies - such as combined convection, film and transpiration cooling - are being developed to reduce heat loads on turbine components thereby improving their durability and efficiency. Secondary air leakage control is also important to turbine efficiency and new methods are under development to minimize the leakages.

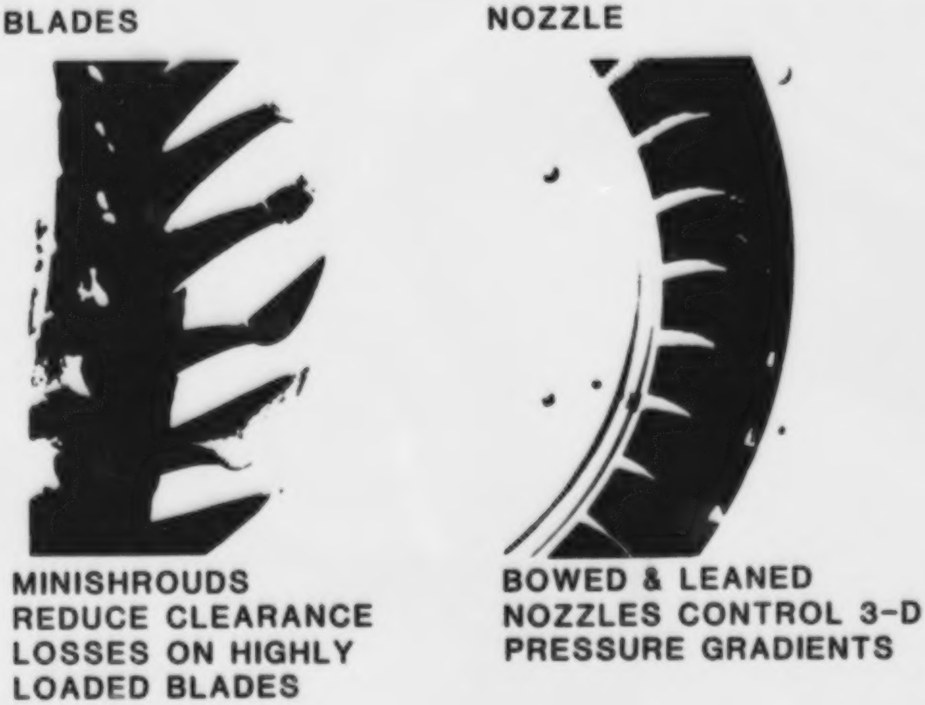
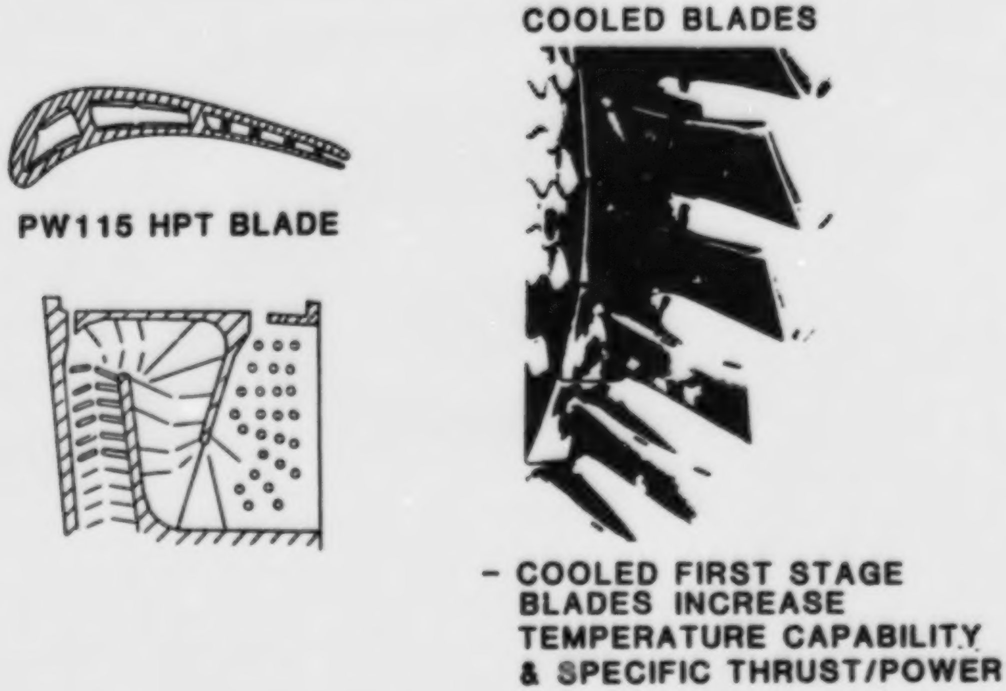

BLADES
NOZZLE
MINISHROUDS REDUCE CLEARANCE LOSSES ON HIGHLY LOADED BLADES
BOWED & LEANED NOZZLES CONTROL 3-D PRESSURE GRADIENTS

Figure 9 - Three Dimensional Designs of Turbine Airfoils

Combustor Technology: Combustor volumes for small engines, have reduced by over 50% in the last 20 years (Figure 11) and this trend will continue with the objective of reducing combustor size and weight. At the same time demands on combustor life and exit temperature uniformity have increased, as well as for high altitude/cold start light-ups and clean exhaust. High life and longer Time Between Overhaul (TBO) have been achieved through a combination of improved materials and cooling techniques, use of thermal barrier coatings, and use of advanced fuel injection systems such as airblast and pre-vaporizers. Table 4 shows relative advantages of using machined or segmented liner construction or thermal barrier coatings. Figure 12 shows common forms of fuel atomizers; the airblast type having larger passages than simplex pressure atomizers is less prone to contamination. The hybrid type provides for reliable light-ups under adverse conditions and can also be used for dual fuel application, the centre nozzle for liquid fuel and


COOLED BLADES
PW115 HPT BLADE

- COOLED FIRST STAGE BLADES INCREASE TEMPERATURE CAPABILITY & SPECIFIC THRUST/POWER

Figure 10 - Turbine Blade Cooling for Improved Durability

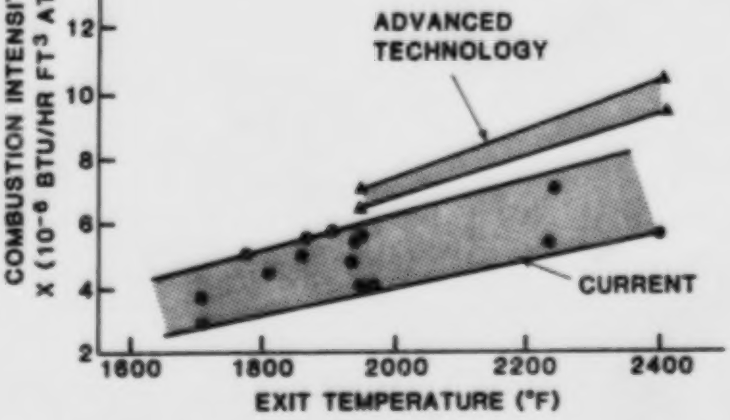
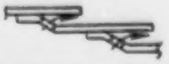
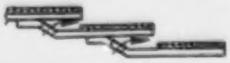
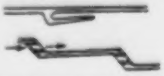
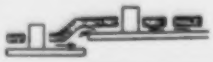
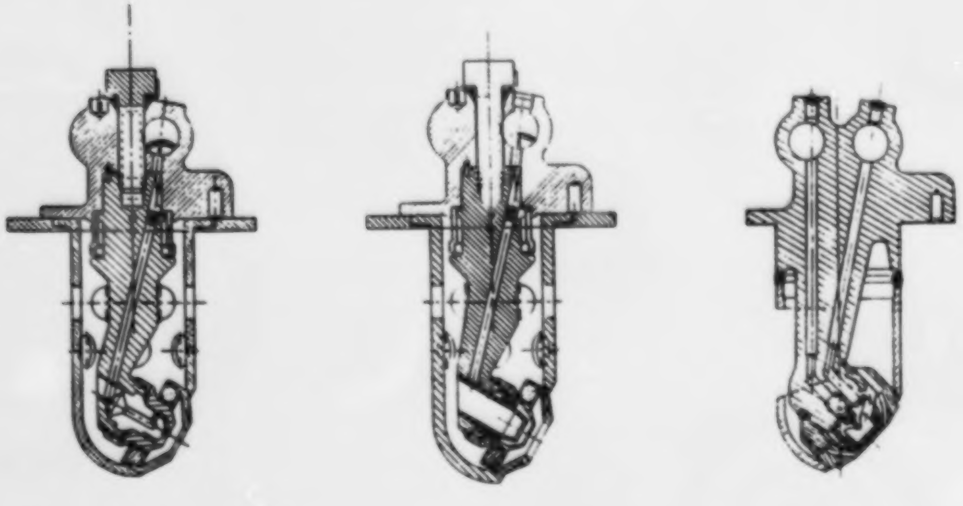


Figure 11 - Trends in Combustion Intensities

outer nozzle for gaseous application. To-date, TBOs exceeding 4000 hours have been demonstrated with airblast and hybrid nozzles. CFD modelling is being used to study the complex flow patterns within the combustor.

Table 4: Effectiveness of Combustor Wall Cooling Methods

CONFIGURATION	REL. DURABILITY
SHEET METAL LOUVER 	1
THERMAL BARRIER COATING 	1.3 - 1.5
MACHINED RING 	1.5 - 2.0
SEGMENTED LOUVER 	15 - 20



PRESSURE ASSIST **AIRBLAST** **HYBRID (DUAL FUEL)**

Figure 12 - Fuel Atomizer Types

Figure 13 shows the grid and CFD model (called "TEACH") results of streamlines within the combustor. Flow path optimization in the initial design stage will significantly reduce combustor development costs. Uniformity of exit

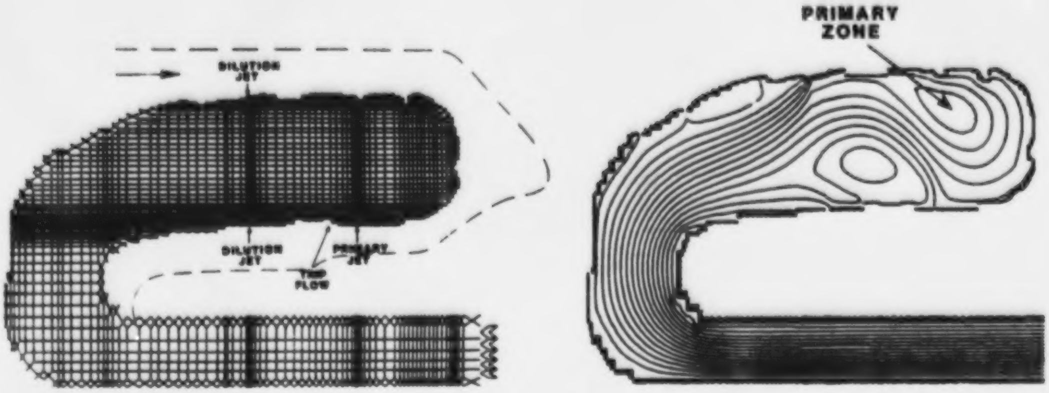


Figure 13 - Modelling Combustor Flows

temperature distribution affects the performance and durability of the rest of engine hot end, and analytical modelling is being increasingly used in temperature profile optimization.

Ability to operate with a wide range of fuels is an important characteristic of gas turbines. The combustor and fuel system may need tuning to accept fuels with different physical and chemical properties and much research has focused on effects of fuel properties on combustor performance and durability. Table 5 shows fuels which have been investigated with PT6 engines, under a recently sponsored program by United States Air Force and Canadian Department of National Defence(3).

Table 5: PWC Investigations with Fuels

Fuel Type	Remarks
Jet A, Jet A-1, JP4, JP5, JP8	Aircraft fuels certificated
Marine Diesel, #2 Diesel, Artic Diesel	Approved for ST6/PT6 use (ambient restrictions)
Aviation Gasoline	Approved for emergency use
Shale Derived JP4 and JP8	USAF/CDND Program, acceptable
Tar Sand Derived Fuels (JP5 type)	USAF/CDND Program - effect metal temperatures, smoke emissions
Broadened Spec. Jet A-1	"
Broadened Spec. JP-4	"
NASA - ERBS Fuel	"

Table 6: Small Engine Exhaust Emission Control Techniques

TECHNIQUE	BENEFITS	DRAWBACKS
a. Lean Primary Zone	Reduces NO_X, Smoke and flame radiation	Increases CO, HC, possible Ignition & Stability degradation
b. Rich Primary Zone	Reduces CO, HC, NO_X improves ignition, stability	Increased Soot, Smoke & flame radiation (hotter walls)
c. Reduced Residence time	Reduced NO_X	Increases CO, HC
d. Increased Residence time	Reduced CO, HC	Increases NO_X
e. Air assist/Air blast atomization	Reduces all emissions & smoke. Reduced flame	Stability & ignition can be impaired
f. Fuel Staging	Reduces CO, HC & NO_X with lean front end	Complexity & possible Pattern Factor penalty
g. Variable Geometry	Reduces all emissions	Complexity
h. Water Injection	Reduces NO_X & Smoke	Increased CO, HC, - system, not for aircraft application, O.K. for Industrial

Combustor flow paths, fuel spray quality and fuel-air mixing in the combustor front end of combustor influence the quantity of visible (smoke) and invisible (CO, HC and NO_X) emissions. Much effort has focused on rendering the exhaust invisible under all operating conditions as well as reducing CO, HC and NO_X emissions. Techniques have been developed to reduce gaseous emissions (Table 6), but the tradeoffs should be carefully considered while incorporating them into specific applications(4).

Materials Technology: Materials technology is influencing developments in all components of small gas turbine engines. While early PT6/ST6 engines used stainless steel for blades, discs and impellers of compressors (Table 2), research with Titanium in the 1960's offered the benefits of lower density, higher strength-to-density ratio and good corrosion resistance. Titanium Ti-6-4 is now the normally used material for compressor components. Higher temperature capability can be obtained with the Titanium alloy Ti-6-2-4-2, which is used in the PW100 high pressure impeller. Further Titanium development to optimize creep and fatigue properties has produced CPW 459, which is a BETA forged Ti-6-2-4-6, and this alloy is being considered for new engines currently under development.

For turbine (hot end) parts, the trend has been for increasing strength and temperature capability, Figure 14. The early PT6/ST6 engines used Inconel 713, a nickel based super-alloy, for turbine blades. Dramatic increases in strength and temperature capability were achieved with directionally solidified alloys (Mar-M-200 DS) introduced in 1974. Further improvements in temperature capability were obtained by use of single crystal alloys.

Single crystal Inconel 792 combines the strength of single crystal material with superior corrosion resistance, and should be particularly useful in marine and industrial applications.

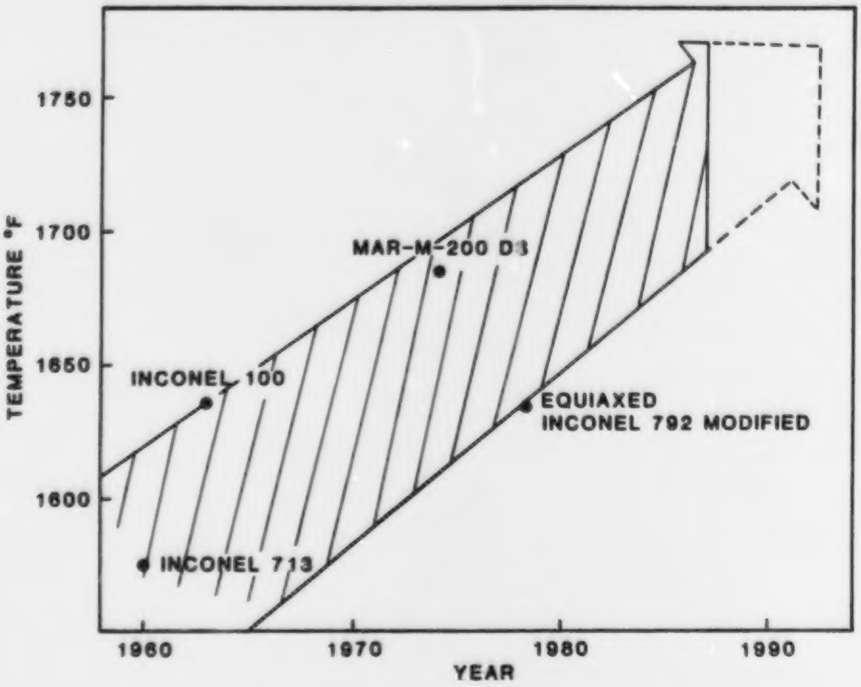


Figure 14 - Trends in Turbine Material Development
(20 KSI, 6000 HR Stress Rupture)

With combustors, Hastalloy X was the standard material until the 1970's when a switch was made to Inconel 625, which offered life improvement as well as better stability and ductility during long term engine exposures. Inco 625 alloy is very corrosion resistant, but life of components can be extended by aluminide coatings, and for regions of high thermal gradients by use of thermal barrier coatings such as Yttria Stabilized Zirconia. Further improvements in thermal capability and oxidation resistance can be obtained with other base materials such as Inco 617 and Haynes 188, which are being evaluated for high temperature combustor applications. For the future, ceramics and ceramic composites offer great potential, but these are still in the research phase, with possible application in the 1990's.

DEVELOPMENT OF HIGH TECHNOLOGY GAS TURBINE ENGINE

The PW100 is a new family of engines by Pratt and Whitney Canada which employs technologies of the eighties. The PT6 started production in 1963 and has been developed for the power range 500 to 1200+HP, with 45 versions certificated. The PW100 series is designed to cover the power range of 1500 to 2600 HP and to date four versions have been certificated.

Table 7: Applications for PW100 Series Engines

MODEL	HP (SLTO)	APPLICATION
PW115, 118, 118A	1500, 1800	Embraer, Brasilea
PW120	2000	De Havilland, Dash 8 - 100
PW120	2000	Aerospatiale ATR 42
PW123	2300	De Havilland, Dash 8 - 300
PW123 AF	2300	Canadair, CL215T
PW124	2400	British Aerospace, ATP
PW124	2400	Fokker, F50
PW124	2400	Aerospatiale, ATR 72

The PW100 engine is specially designed to satisfy the needs of the commercial aircraft market over the next two decades. The main applications are small twin-engine airlines commonly called commuters(5-6). Table 7 shows current applications of the PW100 family. With changes to control system and power turbine aerodynamics the engine is also suitable for helicopter use. Industrial and marine applications for the turbomachine are also being actively pursued.

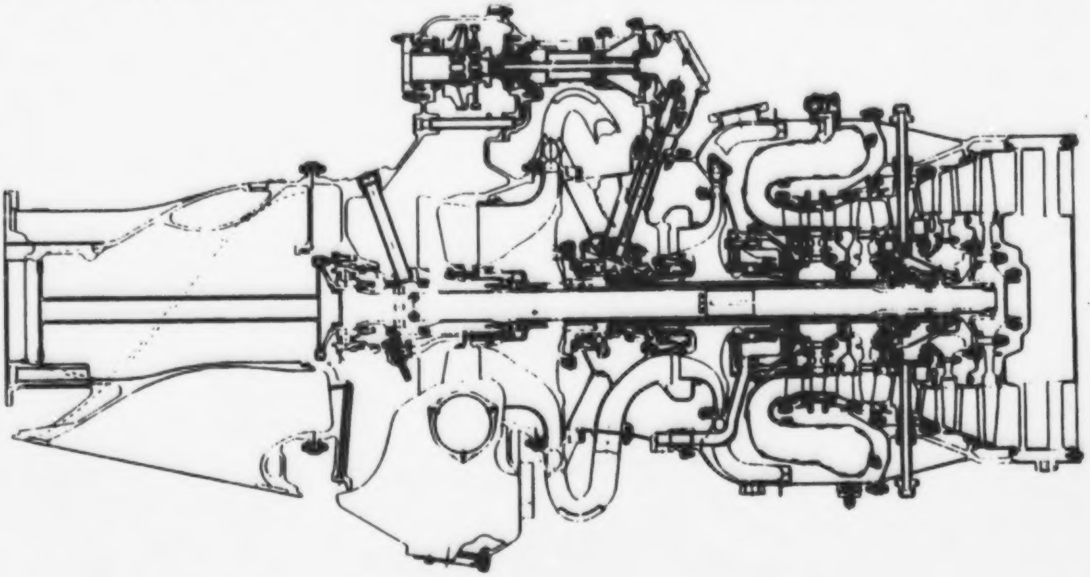


Figure 15 - PW100 Engine Cross-Section

Figure 15 shows the cross-section of the PW100 aircraft power plant. The engine has a two spool centrifugal compressor each spool driven by a single stage turbine. A free turbine provides output power through an offset gear box. Separate drives for the two centrifugal stages provide for each to operate at best speed and efficiency at various power levels without variable geometry or air bleeds. The first stage centrifugal compressor has a transonic inducer, backswept exducer and carefully controlled blade loading; the second high pressure stage has a subsonic inducer and once again carefully matched blade loading. The combustor is of the reverse flow annular type with upstream fuel injection using airblast fuel injectors. The airblast method of fuel injection provides for reduced heat loads to combustor walls, low exhaust emissions/smoke and reduced contamination over long periods (over 4000 hrs.) of operation. Sections of the combustor have thermal barrier coatings made of Yttria-stabilized-Zirconia which keep metal temperatures low and ensure long combustor life. The first stage compressor turbine which drives the HP compressor has segmented cooled vanes, and cooled rotor-airfoil cooling is through relatively large cooling passages to ensure freedom from blockage due to dust. The second stage compressor turbine has cooled vanes but uncooled turbine airfoils. The power turbine is a two-stage uncooled shrouded design for high efficiency at cruise condition.

The PW100 engine has been designed giving utmost importance to reliability (7).Stresses and metal temperatures have been kept deliberately low by present standards and all rotating components have been designed for lives of 30,000 cycles using forged and machined, fracture tough materials. The gearbox is also a rugged design with great attention paid to gear and bearing alignment under all operating conditions.

Operational experience with PW100 since service introduction has been excellent (8). To end of May 1987, 550,000 hours had been accumulated. The high time engine has accumulated over 4000 hours without overhaul compared to industry

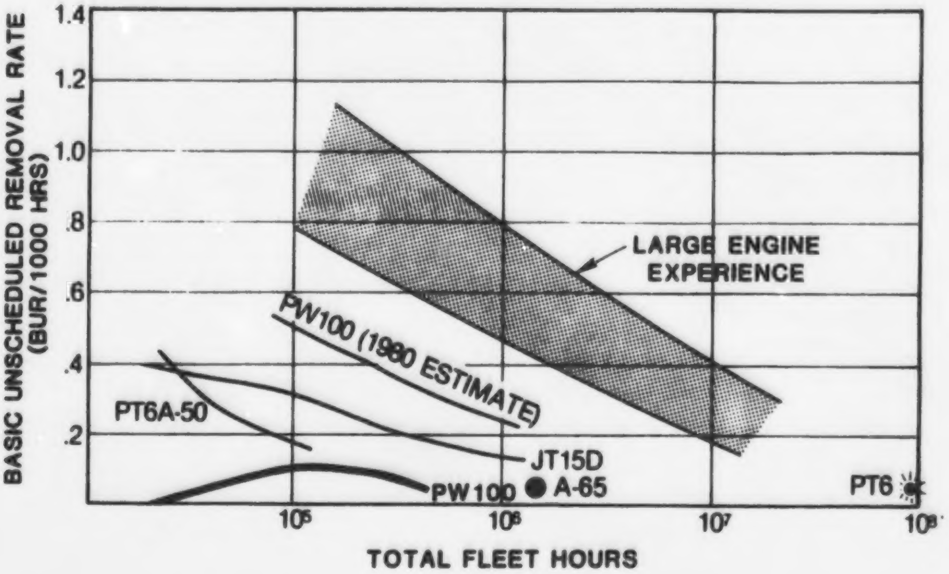


Figure 16 - Engine Reliability Comparison

threshold interval level (TBO) of 3000 hours. The Basic Unscheduled Removal rate (BUR) is currently at 0.025 for 1000 hours of field operation, which compares well with other PWC engines and considerably better than published large engine data (Figure 16). The PW100 field data is from 30 airlines and 8 other operations, with a total fleet of 141 aircraft. This is good testimony to the design philosophy of PW100 focusing on reliability and proven technology.

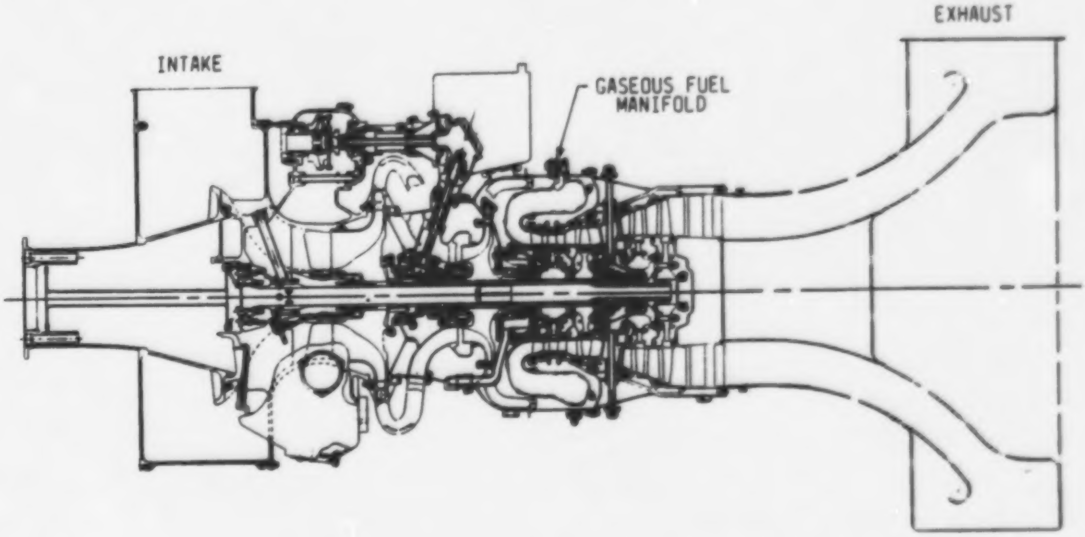


Figure 17 - PW100 Industrial Engine Configuration

There is good potential for industrial application of the PW100 engine (SPW 124-1, Table 1). Figure 17 shows a schematic of an industrial variant proposed for gas pumping application. This maintains all the essentials of the proven turbomachine construction but with reconfigured intake and exhaust. The fuel system can be modified for gas or dual fuel operation. As with the ST6, the free turbine approach provides capability of operating over a range of power shaft speeds at full gasifier power, thus providing a built-in torque converter characteristic. The robustness of the impellers will help sustain good performance in the field because of less susceptibility to erosion. Cooled high turbine blades provide for good life and further durability improvements are possible by use of single crystal blades. The combustor is also a high life component due to use of high performance aerating fuel nozzles and superior aerodynamics as well as mechanical features. The engine can be optimized to meet all the environmental regulations applicable to small industrial engines. Other advantages which would make the aero engine attractive for industrial use are the following:

- Portable power plant which can be packaged into compact installations, due to its low weight and size.
- Good (specific) fuel consumption which is maintained over long periods of operation (Table 1).
- Modular construction which provides for relatively easy maintenance. Access provided for easy engine inspection and condition monitoring.

- High reliability, as evidenced by PW100 field experience.
- Low noise levels in operation and low operator fatigue.
- Resistance to corrosion with additional treatment available as required.
- Tolerance to a wide variety of liquid fuels - light distillates to #2 diesel - and operational capability with gaseous fuels such as natural gas and propane.
- Successful operation over a wide range of altitudes and ambient temperatures (-55°F to +100°F).

CONCLUSIONS

Significant improvements in small aircraft gas turbine component performance have been achieved through advances in modelling, materials and manufacturing methods. Advancements in technology will continue to emphasize fuel burn, size, weight, first cost, operating costs, reliability and ease of installation; much more care being exercised in optimizing engine design to requirements of specific applications. Derivatives of new power plants will be attractive for industrial applications because of their improved reliability, ease of maintenance and operating costs.

Trends in compressor and turbine technology point to efficiency improvements through 3-dimensional (Computational Fluid Dynamic) modelling and greater attention to manufacturing methods. Improvements to materials and cooling techniques can enable operation at higher turbine inlet temperatures. Combustor technology will be toward compact (high intensity) combustors, improved atomization and wall cooling methods for enhancing hot end life, and improving quality of exit temperature distribution. Attention will continue to be paid to engine inlet and exhaust systems as well as reducing secondary air leakages.

The PW100 engine has experienced an excellent service introduction and is proving to be a very reliable power plant. Industrial derivatives for gas pumping and other applications are being proposed which offer portability, ease of installation, reliability, low fuel consumption, tolerance to wide variety of liquid fuels and ability to operate with gaseous fuels.

ACKNOWLEDGEMENTS

The author wishes to acknowledge the technical support for preparation of the paper from Messrs. C. Blizzard, C. Cooper, H.C. Eatock, U. Okapuu, K.V. Patel, G.P. Peterson and R. Sachs. Secretarial help from Mrs. C. Adamo and material preparation by Mrs. I. Tam and Miss P. Chemello are also gratefully appreciated.

REFERENCES

1. Hardy, G., "Applications of the ST6 Gas Turbine Engine", 67069, SAE, p.p. 2278-2294, 1967.

2. Granatelli, A., Cowley, F.D., "Application of Turbine Power at the Indianapolis '500'", SAE Automotive Engineering Congress, 680035, Detroit, Michigan, Jan. 8-12, 1968.

3. Gratton, M., Critchley, I.L., Sampath, P., "Alternate Fuels Combustion Research", USAF Report, AFWAL-TR-2042, 1984.

4. Saintsbury, J.A., Sampath, P., "A Review of Small Gas Turbine Combustor Development", ASME 79-GT-136, 1979.

5. Cook, D.L., "Development of the PW100 Turboprop Engines", 850909, SAE, p.p. 4.740 - 4.746, 1985.

6. Stoten, M.D., Harvey, R.A., "Regional Airline Turboprop Engine Technology", AIAA-83-1158, AIAA/SAE/ASME, 19th Joint Propulsion Conference, 1983.

7. Stoten, M.D., "Design Features of a New Commuter Turboprop Engine", 820717, SAE Commuter Aircraft & Airline Operations Meeting, Georgia, 1982.

8. Peterson, G.P., "PW100 Family - Current Status", CASI Session, Engineering Centennial, Montreal, May 1987.

EMERGING TECHNOLOGIES FOR LIFE CYCLE MANAGEMENT OF TURBINE ENGINE COMPONENTS

A.K. Koul, J-P. Immarigeon and W. Wallace

Structures and Materials Laboratory
National Aeronautical Establishment
National Research Council of Canada
Ottawa, Ontario K1A 0R6

EMERGING TECHNOLOGIES FOR LIFE CYCLE MANAGEMENT OF TURBINE ENGINE COMPONENTS

A.K. Koul, J-P. Immarigeon and W. Wallace

Structures and Materials Laboratory
National Aeronautical Establishment
National Research Council of Canada
Ottawa, Ontario K1A 0R6

Summary

This paper reviews recent developments in the technologies used in gas turbine engine component lifing. Novel nondestructive inspection techniques for microdamage detection in turbine engine components based on small angle neutron scattering, positron annihilation spectroscopy and neutron diffraction are described. A new lifing procedure for low cycle fatigue life limited parts, based on defect tolerance and the application of fracture mechanics principles, is also described. The potential impact of these technologies on component life cycle management and the cost of ownership of turbine engines is discussed.

1.0 INTRODUCTION

During long time service in corrosive environments under high stresses and at high temperatures, turbine engine components such as discs, blades or vanes incur cumulative damage which gradually degrades their mechanical properties and structural integrity. The damage, which can take many forms, depending on component, engine type and service conditions, can have life limiting impact on these parts.

The cost of replacing life-expired components usually represents a significant part of engine ownership cost. This explains the growing interest among operators and manufacturers alike in emerging techniques for damage detection in service-exposed hardware and for residual life predictions that might be used to ensure that the life potential of expensive-to-replace components is fully utilized. In this respect, novel nondestructive inspection (NDI) techniques have emerged that could provide means to assess the extent of microdamage due to creep or low cycle fatigue (LCF) in service-exposed hardware and could be used to predict residual life of service-exposed parts. In addition, a new lifing procedure for LCF life-limited parts has also emerged that relies on combining in-service crack inspections with fracture mechanics principles to establish safe return-to-service intervals for service beyond conventionally established life limits.

It is the purpose of this paper to describe these emerging inspection and lifing technologies and to discuss their potential impact on life cycle management of turbine engine components. In the paper, the conventional lifing procedures used to establish safe life limits for critical engine parts are first briefly reviewed. The forms of life-limiting damage that can occur during service are then described and finally the above mentioned technologies in NDI and component lifing that have the potential to provide more efficient and cost effective use of components are presented and discussed using recent NAE work as case studies.

2.0 LIFING OF GAS TURBINE ENGINE COMPONENTS

A great deal of effort is normally expended by engine designers to establish life limits for components whose deterioration threatens operational safety. The parts that are normally lifed include all rotating hardware such as discs and shafts, with the high pressure turbine disc usually regarded as the most critical part. Other parts that may be lifed include turbine blades and vanes. The lifing procedures employed to establish life limits vary with component and the form of damage. In modern turbines, discs and their spacers are normally designed to withstand LCF and burst by overspeed as well as creep for the hot section side of the engine. Turbine blades and vanes are normally designed to withstand creep as well as thermal-mechanical fatigue and high cycle fatigue. The life limits for these components are established initially based on service estimates of damage accumulation rates. The limits are later subjected to revisions based on field experience. For discs, the most common lifing procedure employed to establish safe life limits follows a time or a "cycle to crack initiation" criterion factored down to represent a minimum property component(1,2). Lately, this approach has been much critized as being overconservative and costly on the grounds that discs are usually discarded with a significant amount of useful residual life(3).

For turbine blades and vanes, life limits may or may not be prescribed by engine designers. In particular, turbine blades and vanes in aero engines are seldom lifed because of the difficulties associated with predicting the service behaviour of metallurgically complex material systems under conditions that can vary widely with user practice. Furthermore, in the case of coated components, these difficulties are compounded due to coating-substrate interactions which are not usually well understood(4). A "life on-condition" approach is sometimes employed where for instance, creep growth or untwist of airfoils is measured and distortion limits are used as retirement criteria(5). However, there are many instances where turbine blades and vanes are used without any regard for creep, thermal fatigue or other forms of damage accumulation. In such cases, the components are used potentially until failure occurs(5).

3.0 COMPONENT DEGRADATION IN GAS TURBINE ENGINES

A listing of potential failures modes for turbine engine components is given in Table I for land based and aero or marine propulsion turbines. The life limiting modes differ somewhat with the type of turbine and its application. In this respect, there are perhaps two major differences between operating conditions for industrial land based turbines and military aircraft or marine gas turbines. Firstly, land based power plants tend to operate for long periods of time under relatively stable conditions and therefore problems due to thermal fatigue are usually less severe. Secondly, land based turbines are often operated on relatively clean fuel and in salt free atmosphere and therefore problems arising from hot corrosion tend to be less severe. However, it is difficult to generalize as these problems are in reality not unknown to land based turbines particularly when they run on low quality fuel with high sulfur content or when they operate in corrosive marine environments. Also, there are situations where land based turbines are repeatedly started at short intervals as for instance in peak load generator drives in which cases thermal fatigue can prove life limiting for hot parts.

During service, components such as discs, blades and vanes can incur both surface and internal damage. Surface damage due to erosion, corrosion, oxidation, wear, fretting or impact can promote crack nucleation at stress concentration sites associated with the

Table 1: Degradation Modes and Life Limiting Properties for Turbine Engine Components

COMPONENT	LAND BASED TURBINES	AERO ENGINE TURBINES
COMPRESSOR		
BLADES	ER - COR, HCF	ER - COR, HCF
VANES	ER - COR	ER - COR
DISCS	LCF	LCF
TURBINE		
BLADES	C, HC	TF, C, HC
VANES	HC, C	TF, HC, C
DISCS	C, LCF	LCF, C

ER : EROSION
HCF : HIGH CYCLE FATIGUE
TF : THERMAL FATIGUE
HC : HOT CORROSION
COR : CORROSION
LCF : LOW CYCLE FATIGUE
C : CREEP

damaged areas. These cracks can then propagate under the influence of centrifugal and/or cyclic (mechanical or thermal) loads during service and eventually cause component failure. Meanwhile, internal damage is related to microstructural changes that occur slowly during service exposures at a rate which is strongly influenced by service temperature and stresses and therefore, to a great extent, by user practice. These microstructural changes can be associated partly with plastic damage accumulation and partly with metallurgical ageing. Depending on component, alloy composition and service conditions, the plastic damage may be due to creep, LCF or creep-fatigue interactions, while ageing may include coarsening of precipitates (sometimes stress-assisted), carbide reactions and precipitation of embrittling phases(6,7). As the internal damage builds up, the resistance of the components to deformation under static (creep) or cyclic (LCF) loading is reduced. Extensive loss of resistance to plastic deformation can lead to cavititation and internal cracking under creep conditions(6) or ductility exhaustion and crack initiation under LCF or thermal fatigue (TF) conditions(7), which ultimately leads to failure in either case. Examples of the microstructural changes associated with these modes of internal damage are presented below.

3.1 Creep Damage

Creep is usually predominant in mid airfoil sections of turbine blades and in some cases in rims of turbine discs where the stresses and temperatures may be sufficiently high to cause time dependent plastic deformation. The component service temperatures are usually high enough to cause ageing of the material which results in loss of creep strength and therefore its load bearing capacity. An example of ageing in an IN738 blade from a land based industrial turbine is shown in Figure 1 which compares the morphology and distribution of the γ' strengthening phase in new and service exposed conditions(8).

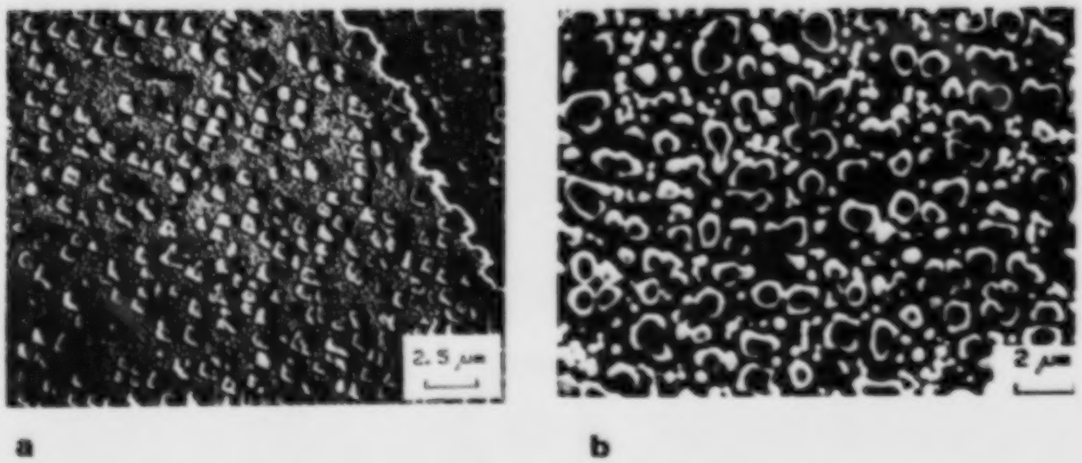


Figure 1 Typical microstructures of IN 738LC - a blade alloy used extensively in industrial and aero engines, a) in the new, as-heat treated condition exhibiting distributions of γ' and b) after 30,000 hours of service showing that the effects of ageing on γ' precipitate morphology and distribution are quite significant (after Koul and Castillo, Ref. 8).

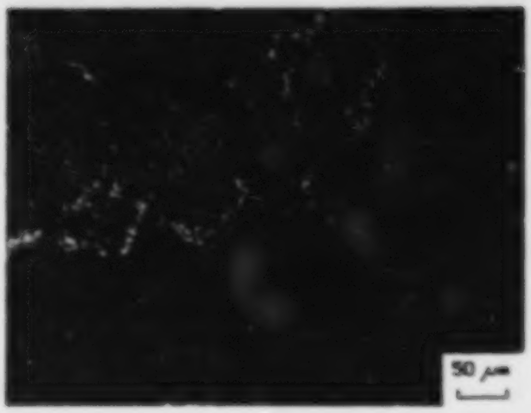


Figure 2 Example of creep voids along the grain boundaries transverse to the blade axis of an Alloy 713C blade from an aero engine (after Chow et al, Ref. 5).

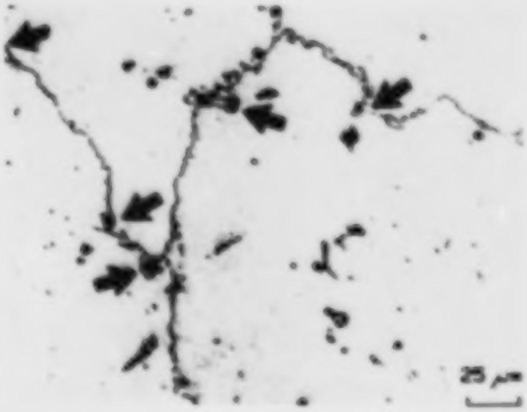


Figure 3 Evidence of carbide reaction at and near the grain boundaries in a service-exposed IN 738LC blade. Arrows point to $M_{23}C_6$ carbides produced by degeneration of the primary MC carbides (after Castillo and Koul, Ref. 9).

After long time service, the spheroidal secondary γ' has virtually disappeared whereas the primary γ' has coarsened extensively and changed from a cuboidal to a rosette morphology.

The loss of matrix strength due to γ' coarsening leads to stress concentrations at the grain boundaries which in turn promotes cavitation and internal cracking(5), Figure 2. Cracking is further facilitated by carbide reactions at or near the grain boundaries, Figure 3, which causes embrittlement of the grain boundaries(9) or by the precipitation of needle like phases (σ or μ) which embrittles the grain interiors(5), Figure 4.

3.2 LCF Damage

LCF damage is dominant in bores and bolt hole areas of compressor and turbine discs. It is caused by stress cycling associated with engine start-up and shut-downs and with RPM excursions during service. Microstructurally, LCF damage manifests itself in the form of intense slip banding, microtwinning and dislocation activity as shown in Figure 5 which compares the microstructure of the bore regions of a new and a service-exposed A286 turbine disc(10). LCF damage can also lead to precipitation of embrittling phases such as eta in A286 or to coarsening of strengthening phases such as $M_{23}C_6$ in AM355 compressor discs(10,11). While the driving force for LCF damage accumulation in compressor discs is primarily mechanical, in hot section areas of the engine it may also be associated with thermal cycling. Thermal fatigue (TF) damage in vanes and high performance blades is often a life limiting factor in modern turbines.

3.3 Macro Damage in Blades, Vanes and Discs

The forms of internal damage described above are precursors to component cracking. For many critical parts, the time spent while accumulating internal damage prior to the formation of a detectable crack (using conventional NDI techniques) usually represents a significant fraction of total component life. This is particularly the case with compressor and turbine discs and turbine blades and vanes. In these components, internal damage builds up slowly, usually in a highly localized manner, for instance near design discontinuities such as bolt holes in discs or cooling holes in blades or at hot spots in turbine blades and vanes, until cracking occurs. Examples of crack nucleation due to LCF and TF are shown in Figure 6 and 7 for bolt hole cracks in a compressor disc(10) and for a cooling hole crack in a turbine blade(12), respectively. Cracking in these examples was the cause for component rejection.

4.0 NOVEL NDI TECHNIQUES FOR MICRODAMAGE DETECTION

While surface damage and surface connected cracks can be detected within limits by conventional NDI procedures such as those based on liquid penetrant, eddy currents or ultrasonic inspection, internal damage can only be reliably assessed destructively either by microscopy or mechanical testing. It would be highly useful to be able to quantify internal damage of critical components in a nondestructive way, particularly if at the same time the internal damage could be correlated with loss of mechanical properties. This capability would help operators and overhaulers in assessing the serviceability of service-exposed parts and in predicting their residual lives.

Recently, a number of novel NDI techniques have emerged that appear promising for detecting microdamage in creep and LCF life limited components. These techniques, which include small angle neutron scattering (SANS), positron annihilation spectroscopy (PAS) and neutron diffraction (ND), have already been explored and shown to be capable of revealing plastic damage and ageing effects in turbine engine materials(13-16). A brief description of their working principles and capabilities is given below.

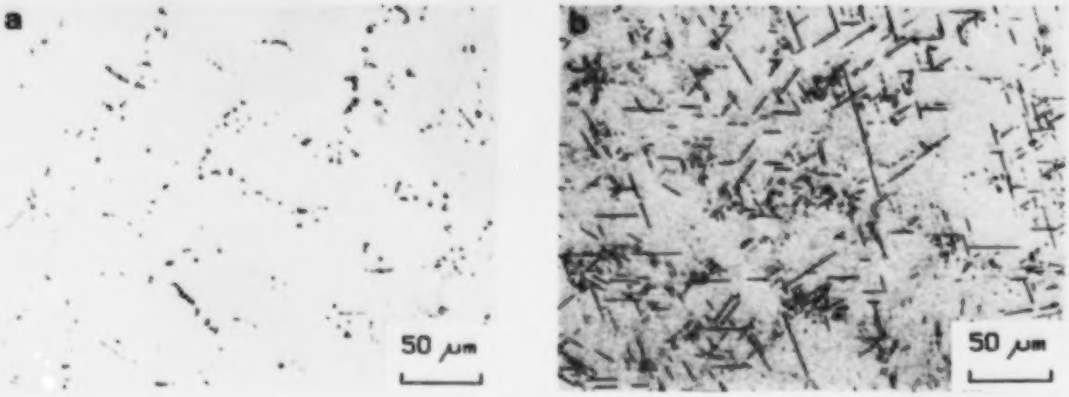


Figure 4 Mid airfoil microstructures in an Alloy 713C blade showing the effects of service-exposure on minor phase precipitation, a) new blade microstructure showing primary MC carbides in a blocky morphology, b) after long time service showing profuse precipitation of a needle like phase (σ) in the grain interiors (after Chow et al, Ref. 5).

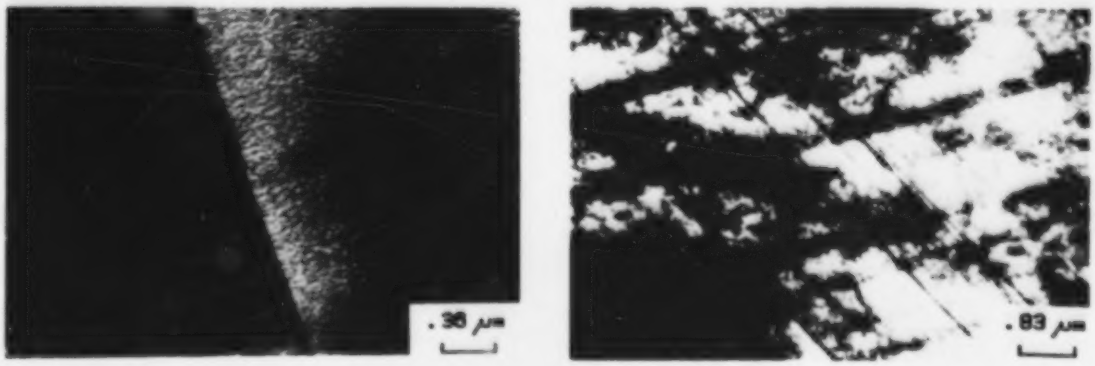


Figure 5 a) as-heat treated and b) service-exposed, microstructures in the bore of a A286 385 CAN 40 turbine disc showing evidence of intense planar slip, microtwinning and high dislocation densities in the service-exposed part (after Thamburaj et al, Ref. 10).

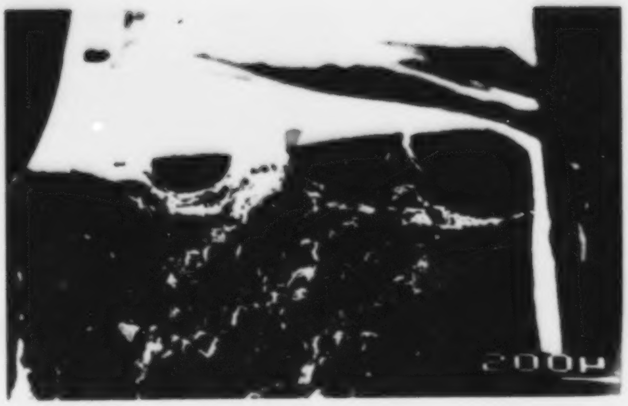


Figure 6   Example of multiple crack initiation in an assembly bolt hole of an aero engine compressor disc (after Thamburaj et al, Ref. 10).

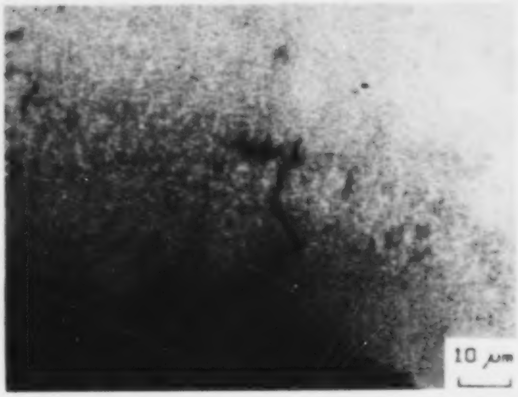


Figure 7   Evidence of a thermal fatigue crack in a high performance internally cooled turbine blade emanating from the protective coating at a leading edge cooling hole orifice (after Bellinger and Koul, Ref. 12).

4.1 Small Angle Neutron Scattering (SANS)

SANS is a technique that relies on detecting the contrast in neutron scattering amplitude arising from differences in atomic or molecular aggregation in a given material(13,14). The technique requires a neutron source (nuclear reactor or particle accelerator) capable of generating neutrons with long wavelength in the range of 4 to 20Å and a position sensitive detector that can detect scattering intensity for scattering angle below 5°. A typical schematic diagram of the SANS experiment is shown in Figure 8(13).

The SANS technique could conceivably be used to detect bulk defects such as creep voids in the size range of 0.01 μm to 0.1 μm and above, and bulk material changes such as γ' coarsening in thick (several mm) materials. It is not certain whether SANS can be successfully used to determine the void distribution in the presence of second phase precipitates such as primary (cuboidal) γ', secondary (spherical) γ' and grain boundary $M_{23}C_6$ and M_6C carbides because of their mutual interaction effects on neutron scattering. However, based on new blade data, it should be possible to separate neutron scattering caused by voids from that caused by second phase precipitates. Figure 9 shows creep void distributions observed in creep tested specimens of an Fe-base superalloy (Alloy 800) using SANS technique(13). Note the technique is capable of detecting changes in creep void distributions in 5% and 10% crept specimens.

4.2 Positron Annihilation Spectroscopy

PAS is a technique that relies on the attractive interaction of positrons with material void-like defects such as vacancies, microvoids and dislocations(15). The eventual fate of a positron is to undergo annihilation with an electron yielding two γ-ray

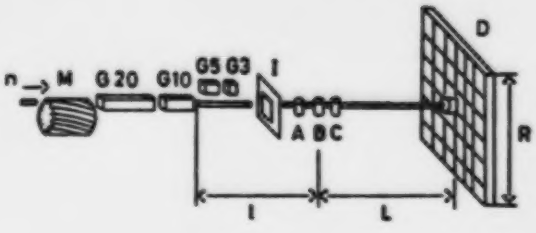

n
M
G 20
G10
G5 G3
I
A B C
D
R
l
L

Figure 8 A schematic diagram of a SANS experiment where A, B, C indicate different sampling locations possible; D is the area sensitive detector; I represents the variable diaphram; M indicates the drum of the mechanical slot selector and G3, G5, G10, G20 are movable neutron guides that allow collimation length adjustments. (After Nilsson and Roth, Ref. 13)

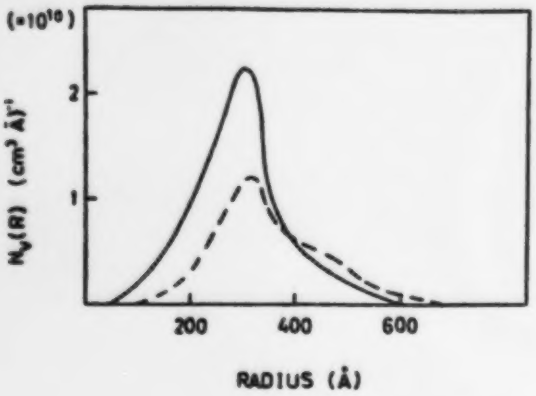


Figure 9 Radius distribution functions of creep voids in Alloy 800 samples; solid line shows void distribution after 5% creep whereas the dotted line indicates a shift in the void distribution after 10% creep (after Nilsson and Roth, Ref. 13).

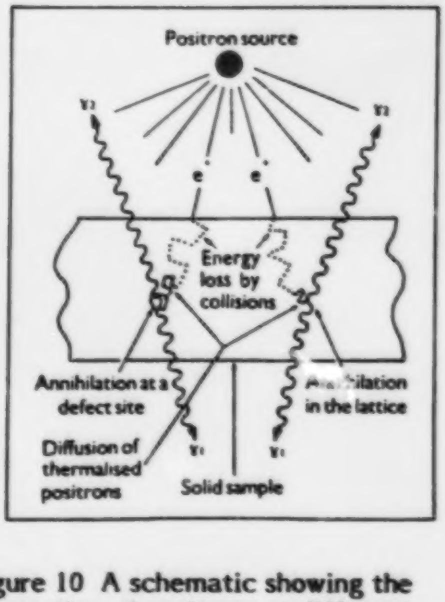

Positron source
γ2
γ2
e+
e+
Energy loss by collisions
Annihilation at a defect site
Annihilation in the lattice
Diffusion of thermalised positrons
γ1
γ1
Solid sample

Figure 10 A schematic showing the interaction of positrons at different sites within a solid sample (after Smith, Ref. 15)

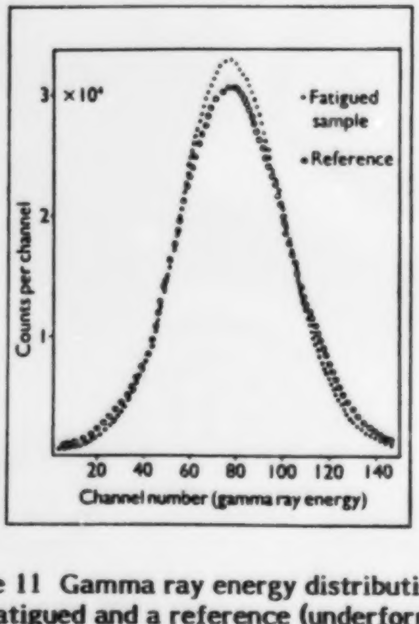


Figure 11 Gamma ray energy distribution of a fatigued and a reference (underformed) Ti specimen using PAS (after Smith, Ref. 15)

photons. The rate of annihilation of the positrons and the detailed characteristics of the energy and momentum distributions of the photons reflect the characteristic properties (density, momentum and velocity distributions) of the electrons with which the positrons annihilate; since these characteristic electronic properties differ, in turn, between the perfect lattice and at defects in the lattice. Measurement of the detailed annihilation characteristics of a positron population can, in principle, offer information about the concentrations and types of defects present(18). Conventional positron sources are radio isotopes and are produced in cyclotrons. The positron sources most commonly employed are 22Na and 68Ge. These isotopes are commercially available from numerous distributors and can be obtained in liquid or in a wide variety of solid shapes such as foils or powders. In practice, the positron source is located at the surface of the part to be analyzed and a high resolution photon energy detector is used to detect the emitted photons. A schematic diagram indicating the interaction of positrons with solids is shown in Figure 10(15).

PAS is sensitive to void sizes ($\ll$0.01 μm) which cannot be easily resolved by SANS and in particular, it is sensitive to defect concentrations of 1 part in 10^7.

It is, however, primarily a near surface detection technique with penetration potential typically of the order of 10^{-4} to 10^{-7} m depending on the type of material under investigation. Higher penetration, up to 10^{-2} m, are however possible using higher energy positron beams(18). Whether the defect density information that might be derived from PAS can be directly correlated with the degeneration of specific high temperature properties can only be verified through well controlled experiments. It is conceivable that the initial stages of LCF damage due to either mechanical or thermal cyclic loads can be monitored prior to the onset of crack initiation by this technique. Figure 11 illustrates the differences in the γ-ray photon counts of a fatigued and a reference (undeformed) Ti sample(15).

4.3 Neutron Diffraction (ND)

ND is a technique that relies on diffraction of a neutron beam by the atomic planes of crystalline solids following the same diffraction principles that govern X-ray diffraction (Bragg's Law)(16,17). However, much greater penetration into materials can be achieved with neutrons which means that measurements can be made several centimeters below the surface of a metallic part. Also, by careful adjustment of incident and diffracted beams, it is possible to achieve selected volume diffraction and therefore, diffraction analysis of a specific location deep below the surface of a part is entirely feasible.

As in SANS, the ND based inspection requires a neutron source of specific wavelength and ancilliary detection equipment. Facilities for both SANS and ND inspection exist in Canada and are located at the Chalk River Nuclear Research Laboratory of AECL and at McMaster University for SANS analysis. A schematic showing the neutron spectrometer facility available at AECL is presented in Figure 12(17).

Neutron diffraction can provide information on residual stresses and on crystallographic texture. The presence of residual stresses causes the interplanar spacing of atoms to change and produces a detectable shift in diffraction angles. This shift can be used to calculate the strains and therefore the stresses. Neutron diffraction can also be used to identify and quantify minor phases in multiphase alloys. This is done by measuring position and intensity of the major diffraction peaks that uniquely identify the atomic structures of the minor phases, again following the principles of X-ray diffraction.

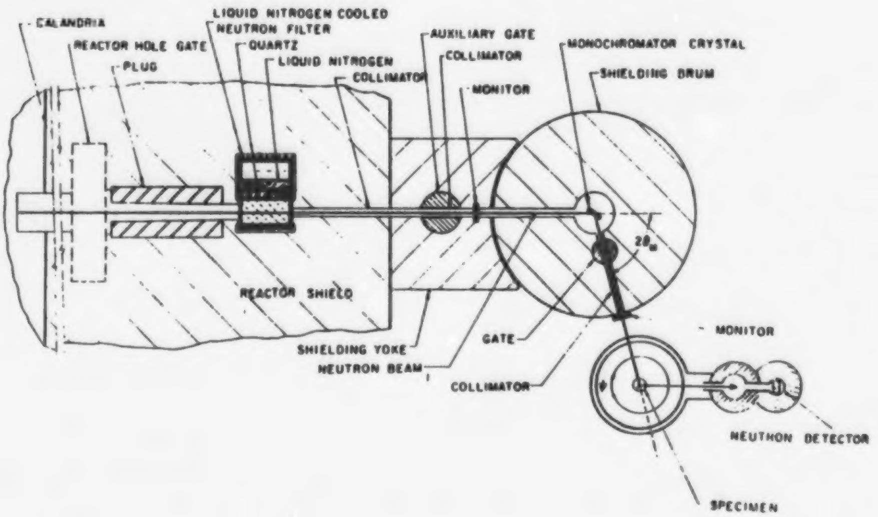

CALANDRIA
REACTOR HOLE GATE
PLUG
LIQUID NITROGEN COOLED NEUTRON FILTER
QUARTZ
LIQUID NITROGEN
COLLIMATOR
AUXILIARY GATE
COLLIMATOR
MONITOR
MONOCHROMATOR CRYSTAL
SHIELDING DRUM
$2\theta_M$
REACTOR SHIELD
MONITOR
GATE
SHIELDING YOKE
NEUTRON BEAM
COLLIMATOR
NEUTRON DETECTOR
SPECIMEN

Figure 12 A schematic of neutron spectrometer facility available at AECL, Chalk River (after Holden et al., Ref. 17).

A ND based inspection technique for turbine engine components could conceivably be used to monitor LCF damage accumulation by following the line shift and/or the line broadening of the prominent diffraction peaks. Such line shift and broadening may be expected from plastic damage accumulation and associated residual elastic strains. The ND technique might also be used to monitor the service-induced changes in the constitution and volume fraction of MC, γ', $M_{23}C_6$ and M_6C in nickel base superalloy hot section components. The selected volume ND technique could be used to monitor the phase changes in selected areas within the leading or the trailing edge of turbine blades or vanes or across mid airfoil section where damage may be expected to be more prominent. Furthermore, texture analysis might be used to reveal service-induced rafting of the γ' phase in directionally solidified and/or single crystal blades.

4.4 State-of-the-art

None of these inspection techniques have been developed to the point where they can be routinely applied to monitor microdamage accumulation in turbine engine components. Work is underway in many laboratories worldwide including Canada to assess their sensitivity and reliability. Preliminary studies conducted jointly by NAE and AECL have shown that the techniques are sensitive to the presence of microstructural variations in turbine blades(19,20). A programme is currently being planned at NAE to calibrate the techniques with respect to creep and LCF damage using laboratory test coupons exposed to simulated service conditions. It must also be emphasized that the three techniques should not be viewed as procedures that compete with one another to generate information on internal microdamage. Instead, they are complementary in the sense that they can each provide different information on microdamage which, brought together, could be used to assess serviceability of used components.

5.0 DAMAGE TOLERANCE BASED LIFING FOR TURBINE ENGINE COMPONENTS

As indicated earlier, rotating parts such as compressor and turbine discs are usually life-limited due to LCF damage accumulation. A time is ultimately reached in the life of these components when they are no longer capable of meeting minimum design requirements and are therefore considered unsafe and retired. Life limits are established for these parts by engine designers with a view to minimizing the risks of in-service failures and also to ensure operational safety. The most common lifing method employed for these parts follows a "cycles-to-crack-initiation" criterion where a minimum life capability is defined statistically for simulated service conditions through extensive coupon testing and component test verification in spin pit test rigs. The statistical minimum is usually based on the probability that 1 in 1000 components will have developed a detectable crack (typically chosen as a crack 0.8 mm long). For aero engine discs, in civil aircraft applications, the method used to establish the statistical minimum is sanctioned by aviation regulatory bodies such as the Federal Aviation Administration. For new engines, the initial allowable life is usually reduced to one third of the minimum life capability and, as field experience accumulates, life extension towards the minimum life capability is eventually granted.

5.1 Problems with Conventional Lifing

There are two major concerns when a cycle-to-crack-initiation rejection criterion is used to life rotating parts. The first one is that, by implication, 99.9 percent of the components will be retired before any detectable crack has formed and it is possible that these parts may be quite capable of significantly longer service before they develop a 0.8 mm crack. A study conducted by Pratt and Whitney (Florida) has revealed that the residual life potential for F100 rotors can be quite significant, up to 10 lifetimes for over 80% of the retired parts as shown in Figure 13(3). Secondly, the components may be capable of tolerating crack sizes much greater than 0.8mm mentioned above. The limit actually reflects the sensitivity and reliability of current NDI methods rather than the mechanical tolerance of a particular part for cracks. During inspection any crack must be detected with high reliability, usually expressed as probability of detection (POD) at high confidence, and requirement is usually expressed as 90% POD at 95% confidence. The more sensitive is the NDI method, the smaller is the expected flaw at the 90/95 limit.

If component lives could be set based on a reliable knowledge of critical flaw sizes and the time or number of operational cycle required for a flaw to grow from its size at the detection limit to its critical size, then lower operating costs might be expected. The potential cost saving that could be derived from an alternative rejection criterion based for instance on some measure of crack propagation life is clearly indicated in Figure 14, (taken from a study by 'General Electric')(1).

5.2 Alternatives to Conventional Lifing

From a users point of view the economics of retiring expensive-to-replace hardware at some generic minimum life based on initiation criteria is obviously not very attractive. To ensure that the life potential of rotating hardware is fully utilized alternative lifing procedures, based on defect tolerance and the application of fracture mechanics principles, have been suggested. The philosophy behind these alternative lifing procedures assumes that components contain inherent defects that act as propagating cracks(2,3,7,11) and that components may be capable of continued safe operation during crack growth, providing the cracks grow sufficiently slowly during service to allow their growth to be reliably monitored through regularly scheduled inspection.

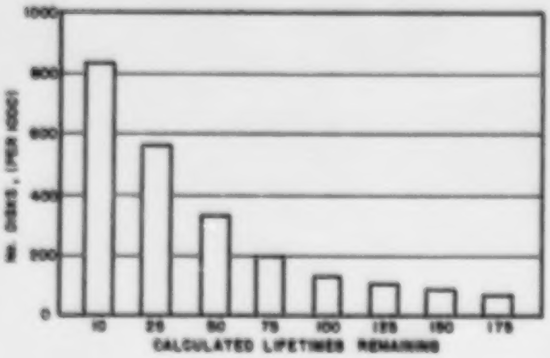


Figure 13 Distribution of calculated residual lives for retired F100 rotors showing that the majority of discs have useful life after retirement when a conventional crack initiation lifing criterion is used to life the components (after Annis et al, Ref. 3).

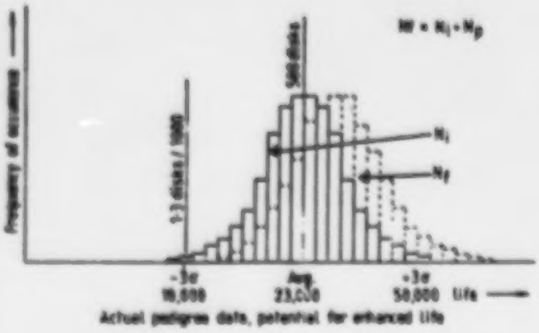


Figure 14 Schematic field life distributions showing the potential benefits of using a rejection criterion based on some measure of crack propagation rather than conventional crack initiation criterion (after Coles, Ref. 1).

In their most elementary forms, these alternative damage tolerance based lifing procedures, which are also known as Life On Condition, Retirement for Cause or simply fracture mechanics lifing, assume that the fracture critical locations of a component contain cracks of a size that lie just below the detection limit of the NDI technique used to inspect the component. The crack is then assumed to grow during service in a manner that can be predicted by linear elastic fracture mechanics, or other acceptable methods, until a predetermined dysfunction limit is reached beyond which the risk of failure due to rapid crack growth becomes excessive.

The rates of crack growth and the dysfunction crack sizes are established analytically based on best estimates of service loads and material properties. The time or number of cycles required to grow the assumed crack to its dysfunction size is then used to define a safe inspection interval (SII) usually by dividing the life to dysfunction by a safety factor of 2(11). This life cycle management concept is illustrated schematically in Figure 15 which shows that at the end of one SII, crack free components are returned to service for another SII and the procedure repeated until a crack is found. In this manner, components are retired on an individual basis when their condition dictates.

The above procedure (also known as deterministic fracture mechanics (DFM) approach) makes worst case assumptions for all parameters used and can therefore result in a conservative estimate of the SII. It has been suggested that more realistic estimates of SII are possible using probabilistic fracture mechanics (PFM) principles. In these alternative procedures, the probability distributions associated with the occurrence of initial flaw sizes, crack detection, crack growth rates and dysfunction crack sizes are used to establish a SII distribution from which an inspection interval can be selected that ensures a sufficiently low but cost effective failure probability(10,11,21). The predictions of either approach can be further revised on the basis of field experience and this is known as combined analysis (CA)(10,11).

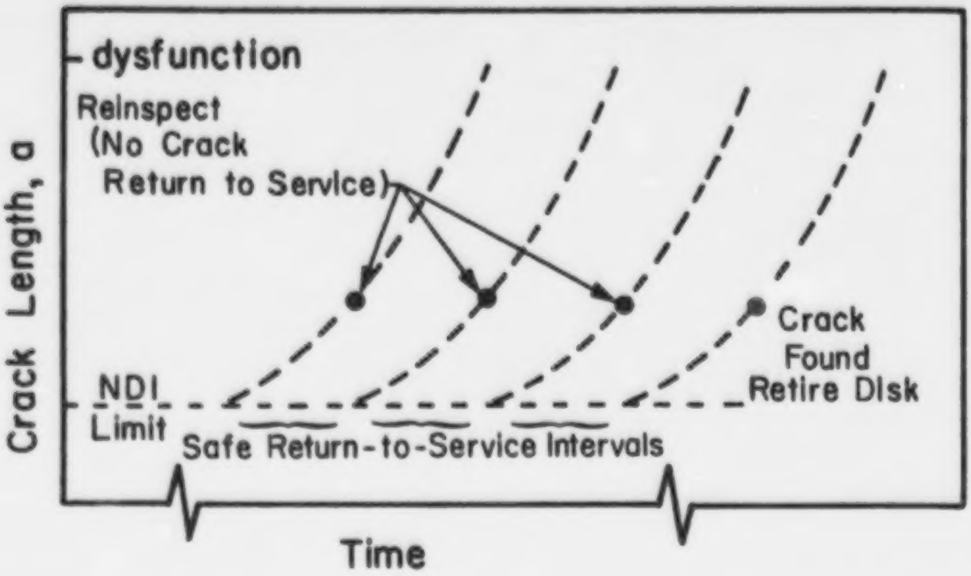


Figure 15 Schematic representation of the damage tolerance based life cycle management of a component where fracture mechanics is used to establish a safe return to service interval until a dysfunction crack is found and the part is retired.

5.3 Applications

The cost benefits implicit in fracture mechanics based lifing may be considerable as the full life potential of expensive to replace components can be utilized. Other benefits may also accrue in terms of engine reliability and safety. For instance, if unforeseen circumstances such as corrosion or fretting cause early cracking it can be argued that a lifing procedure based upon inspection related initial crack size offers a safe approach(2) primarily because operators/overhaulers will be looking for cracks rather than relying only on a prescribed conventional life limit.

These damage tolerance based lifing procedures are currently the subject of much discussion and are receiving attention both for establishing component design life limits in new engines and for extending the usable service lives of discs in conventionally lifed engines. The concepts have already been extended for designing several new generation engines, including F100-PW-220, F110-GE-100 and F109-GA-100 engines. They have also been applied to existing engines such as PW F100, GE TF34 in the US and to GE J85-CAN40/15 in Canada for extending the usable service lives of discs. The experience gained at NAE in damage tolerance based lifing and life extension of J85 components is currently being expanded for possible application to other Canadian Forces engines including the F404 of the new CF180 fighter.

In spite of these applications, it is widely recognized that damage tolerance based lifing has not yet emerged as a fully mature technology and that substantially more work needs to be done to understand all associated implications. In particular, supporting technologies need to be assessed and perhaps further developed if the concepts are to be applied reliabily. These supporting technologies include nondestructive inspection, mechanical testing of test coupons and components, structural analysis, mission profile analysis and condition monitoring of components.

The Structures and Materials Laboratory of NAE and its industrial and university partners have on-going activities in each of these supporting technologies using Canadian Forces components as development tools. Work is underway at NAE to generate statistically significant data bases on the sensitivity and POD of LCF cracks emanating from disc bolt holes for a wide variety of NDI techniques including liquid penetrant inspection (LPI), eddy current inspection (ECI) and ultrasonic C-scan techniques. The results of this work have demonstrated that ECI and C-scan techniques have higher sensitivities than LPI but that, although the ECI appears to be superior to C-scan in terms of sensitivity (1 mm for ECI versus 1.5 mm for C-scan at 90% POD level, Fig. 16) ECI generates a larger number of false calls. Consequently, a larger proportion of crack-free components is likely to be rejected when inspected by ECI as compared to C-scan(22).

Work is also underway at NAE to generate room and elevated temperature crack growth rate data for the application of DFM and PFM based life extension concepts to Canadian Forces aero engines. Materials under investigations include AM355 and Ti17 (two compressor disc materials from J85 CAN40/15 and F404 engines, respectively) and A286 and Inconel 718 (two turbine disc materials used in J85 CAN40/15, T56 and F404 engines). Existing test facilities allow crack growth rate testing to be carried out at typical disc operating temperatures of up to 650°C. The effect of service-exposure on LCF crack growth rate for Inconel 718 material using compact tension specimens machined from J85 CAN15 turbine discs is illustrated in Figure 17(23). The data reveal that in high time components CGR is increased by a factor of two to three.

The validity of analytical predictions in crack growth behaviour of actual components is being verified through spin pit testing of actual components, using the University of Toronto test facilities. Spin pit test results for AM355, J85 CAN40/15 fifth stage compressor discs are shown in Figure 18. The AM355 study included constant amplitude and variable amplitude (spectrum) loading as well as quantitative fractographic analysis of selected cracks as a means of comparing field service crack growth rates with those produced during spin pit testing(24). This work relied heavily on mission profile analysis to generate a unique variable amplitude loading sequence for Canadian Forces J85 CAN40 engines(25).

5.4 Potential and Limitations

As pointed out earlier, the damage tolerance based lifing can be employed both at the component design stage or as an in-service life extension scheme to extend the usable lives of discs beyond safe life limits based on crack initiation criteria. In order to apply the concept for life extension, the fracture critical areas of the components must be known and inspectable. Also, statistically significant information must be available on:

a) sensitivity and probability of detection of NDI technique used,

b) operational loads,

c) material crack growth rate data at service temperatures and simulated service environments and

d) dysfunction crack size.

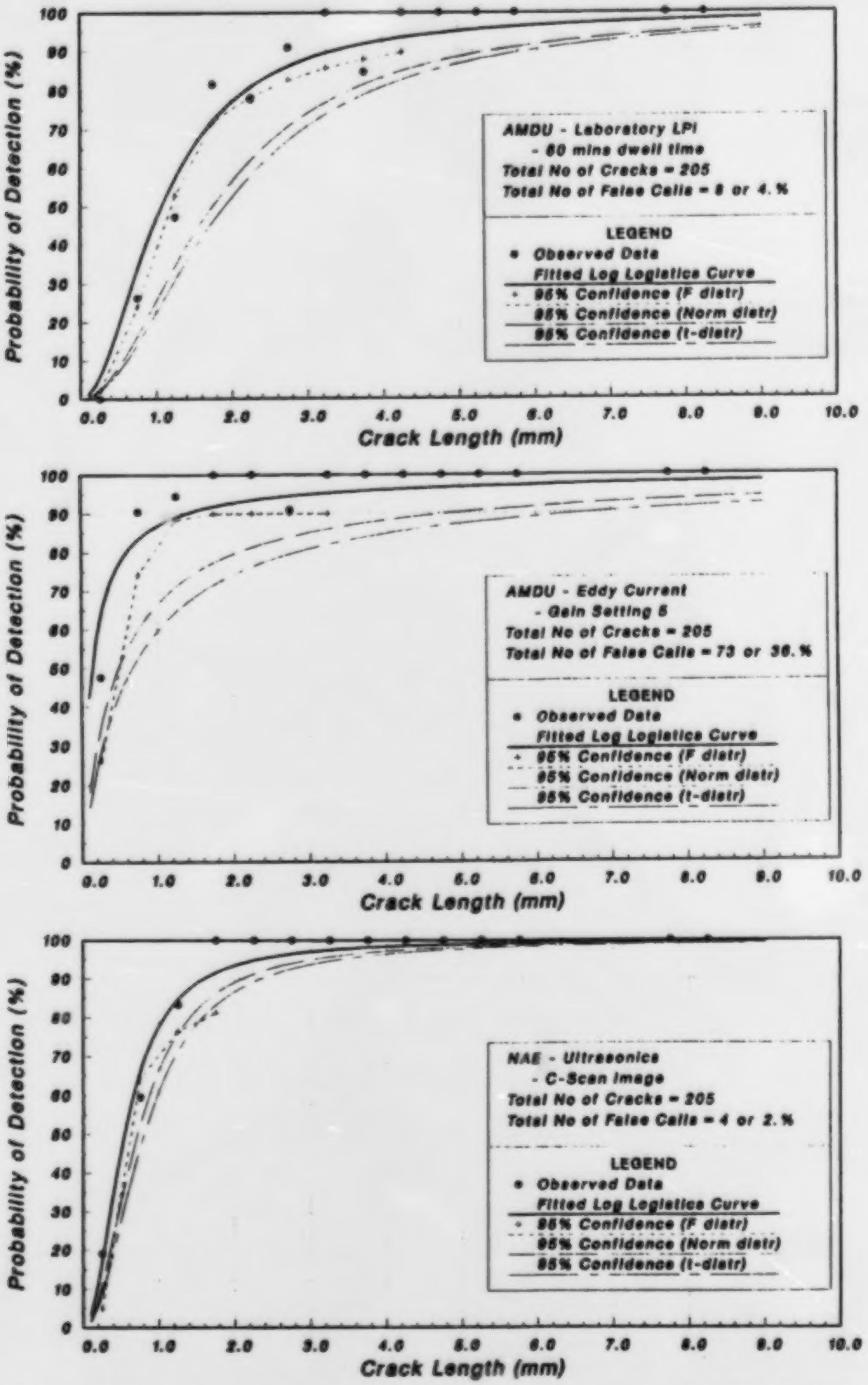


Figure 16 Probability of detection for LCF cracks in compressor disc bolt holes as a function of crack length for a) liquid penetrant inspection, b) eddy current inspection and c) ultrasonic C-scan inspection (after Koul et al, Ref. 22).

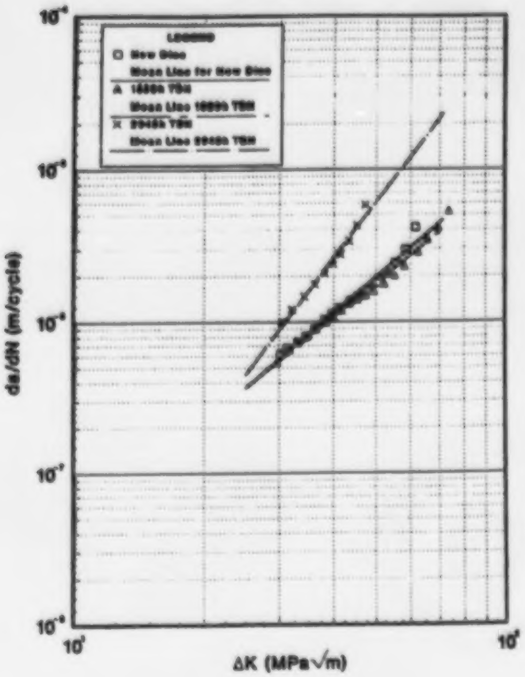


Figure 17 LCF Crack growth rate data for new and service-exposed IN 718 J85 CAN15 turbine discs at 650°C showing that in high time components, the crack growth rate is higher than that for new discs by a factor of two to three (after Pishva, Ref. 23).

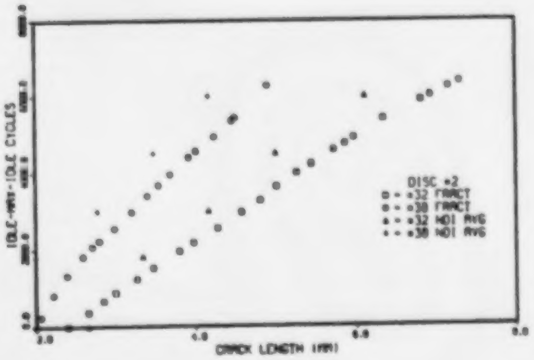


Figure 18 Comparison of crack growth data establish by NDI and fractographic analysis for a J85 CAN40 compressor discs tested under constant amplitude loading in a spin pit (after Hull et al, Ref. 24).

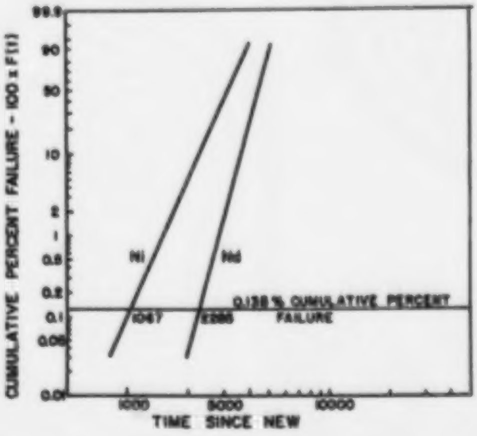


Figure 19 Weibull plots comparing cumulative failure probabilities as a function of service exposure for crack initiation (Ni) and dysfunction crack size (Nd) data bases. At 0.135% failure probability, usable life is increased from 1067 to 2285 hours when applying damage tolerance concepts (after Koul et al, Ref. 11).

This is probably one of the major limitations in applying this concept for post design life extension as the required information is generally not readily available.

This life extension methodology is particularly attractive if original design life limits are dramatically reduced either due to overly optimistic design or increased mission severity. Under such circumstances it may even be cost effective to generate the missing information. In a recent application involving J85 CAN40 compressor discs it was possible to increase the usable service lives of the parts by more than a factor of 2 at 0.135% failure probability as indicated by the Weibull plots of Figure 19. For reasons explained elsewhere(7,10,25), the safe inspection interval in this particular application had been established at 555 hours. It should be pointed out that J85 CAN40 compressor disc is not a good example of the cost effective use of damage tolerance lifing. The safe inspection interval of 555 hours is far too short, and therefore the reliance on very expensive tear down and inspection is too high, for the method to be considered cost effective. For cost effectiveness the SII should be equal to or greater than the established maintenance/inspection interval, which is typically greater than 1600 hours for aeroengines.

It is worth noting that the higher the resolution of the NDI technique employed for inspection, the more cost effective is, in principle, a damage tolerance based lifing methodology. This is because the safe inspection interval increases, since the predicted time to grow a smaller initial crack to dysfunction size is increased. Here, one assumes that short cracks behave in a manner similar to long cracks. However, there is increasing evidence to show that short crack behaviour differs considerably from long crack behaviour and that evaluation of lifetimes based on long crack data can be non-conservative especially at lower stresses(27). The associated risks therefore provide additional incentives for improving the understanding of material behaviour in the presence of defects and modelling techniques to predict material and component behaviour reliably.

6.0 REFERENCES

(1) A. Coles, 'Material Science Considerations for Gas Turbine Engines', Proc. 3rd International Conference on Metals, Vol. 1, Cambridge, England, August 1979, pp. 3-11.

(2) R.H. Jeal, 'Damage Tolerance Concepts for Critical Engine Components', Proc. AGARD-SMP Conference on Damage Tolerance Concepts for Critical Engine Components', AGARD-CP-393, San Antonio, Texas, 1985, pp. I-1 to I-12.

(3) C.G. Annis Jr, M.C. VanWanderham, J.A. Harris Jr and D.L. Sims, 'Gas Turbine Engine Disk Retirement-For-Cause: An Application of Fracture Mechanics and NDE', Trans. ASME, Vol. 103, January 1981, pp. 198-200.

(4) P.C. Patnaik and J-P. Immarigeon, 'High Temperature Protective Coatings for Aero Engine Components', NRC, NAE AN-42, 1986, pp. 1-57.

(5) D. Chow, A.K. Koul, J-P. Immarigeon and D.D. Morphy, 'Service Induced Damage in T56-A7B 1st Stage Turbine Blades - Evaluation and Implications', NRC, NAE LTR-ST-1603, 1987, pp. 1-14.

(6) P.H. Floyd, W. Wallace and J-P. Immarigeon, 'Rejuvenation of Properties in Turbine Engine Hot Section Components by HIP', Proc. Int. Conference Heat Treatment 1981, Birmingham, England, September 1981, pp. 97-104.

(7) A.K. Koul, W. Wallace and R. Thamburaj, 'Problems and Possibilities for the Life Extension in Gas Turbine Components', Proc. AGARD-PEP Conference on Engine Cyclic Durability by Analysis and Testing, AGARD-CP-368, Lisse, The Netherlands, May 1984, pp. 10-1 to 10-31.

(8) A.K. Koul and R. Castillo, 'Assessment of Microstructural Damage and Its Rejuvenation in Turbine Blades', Met. Trans. A, In Press, 1987.

(9) R. Castillo and A.K. Koul, 'Effects of Microstructural Instability on the Creep and Fracture Behaviour of Cast IN738LC Ni-Base Superalloy', Proc. Int. Conference on High Temperature Alloys for Gas Turbines and Other Applications, Eds W. Betz et al, COST 501, Liege, Belgium, October 1986, pp. 1395-1409.

(10) R. Thamburaj, A.K. Koul, W. Wallace and M.C. de Malherbe, 'Life Extension Concepts for Canadian Forces Aircraft Engine Components: J85-CAN40 Compressor and Turbine Materials', NRC, NAE LTR-ST-1585, 1986, pp. 1-186.

(11) A.K. Koul, R. Thamburaj, W. Wallace, M.D. Raizenne and M.C. de Malherbe, 'Practical Experience with Damage Tolerance Based Life Extension Concepts for Turbine Engine Components', Proc. AGARD-SMP Conference on Damage Tolerance Concepts for Critical Engine Components, AGARD-CP-393, San Antonio, Texas, 1985, pp. 23-1 to 23-22.

(12) N. Bellinger and A.K. Koul, 'Coating Degradation and Cracking in a Turbine Blade', NRC-NAE unpublished work, 1987.

(13) G.G. Nilson and M. Roth, 'Small Angle Neutron Scattering Investigation of Voids in Creep Deformed Alloy 800', J. Mat. Sci. and Eng., Vol. 50, 1981, pp. 101-108.

(14) R.J.R. Miller, S. Messoloras, R.J. Stewart and G. Kostroz, 'A Small Angle Neutron Scattering Study of the Temperature and Stress Dependence of the Microstructure of Nimonic Alloy', J. Appl. Cryst., Vol. II, 1978, pp. 583-588.

(15) R.L. Smith, 'Recent Developments in Nondestructive Testing', Metals and Materials Journal, April 1987, pp. 187-191.

(16) S.R. MacEwen, J. Faber Jr. and A.P.L. Turner, 'The Use of Time-of-Flight ND to Study Grain Interaction Stresses', Acta Metall., Vol. 31, No. 5, 1983, pp. 657-676.

(17) T.M. Holden, G. Dolling, S.R. MacEwen, J. Winegar, B.M. Powell and R.A. Holt, 'Applications of Neutron Diffraction to Engineering Problems', Proc. 5th Canadian Conference on NDT, Ed. C.A. Kittmer, Toronto, Canada, October 1984.

(18) G.M. Hood and R.J. Schultz, 'A Study of Deformation of Zr and Ti by PAS', J. Nucl. Mat., Vol. 120, 1984, pp. 94-98.

(19) T.M. Holden, B.M. Powell, S.R. MacEwen, A.K. Koul and J-P. Immarigeon, 'Preliminary Experiments on New and Service Exposed Turbine Blade Using ND', NRC and AECL unpublished work, 1986.

(20) G.M. Hood, S.R. MacEwen, A.K. Koul and J-P. Immarigeon, 'Turbine Blade Characterization by PAS', NRC and AECL unpublished work, 1986.

(21) G.M. Gould and A.K. Koul, 'A Programme for Predicting SII Using PFM Principles', NRC, NAE ST-428, September 1986.

(22) A.K. Koul, A. Fahr, G. Gould and N. Bellinger, 'Importance of Sensitivity and Reliability of NDI Techniques in Damage Tolerance Based Lifing of Turbine Discs', to be presented in AGARD-SMP Workshop on Damage Tolerance Concepts for Critical Engine Components, 1988.

(23) R. Pishva, 'Effect of Rejuvenation Treatments on Fatigue Crack Growth of Service Exposed Turbine Discs', M.Eng. Thesis, Carleton University, 1987 (to be submitted).

(24) A. Hull, R.V. Dainty, M.D. Raizenne and D.W. Hoeppner, 'Fatigue Crack Growth Testing of J85-CAN40 Compressor Discs', CASI Journal, Vol. 32, No. 4, 1986, pp. 321-332.

(25) J.N. Jones and M.D. Raizenne, 'J85-CAN15/40 Engine Mission Profile Analysis', NRC, NAE LTR-ST-1555, November 1985.

(26) A.K. Koul, R. Thamburaj, J. Schofield and W. Wallace, 'Weibull Analysis in Turbine Engine Component Lifing', NRC, NAE LTR-ST-1528, August 1985, pp. 1-24.

(27) T. Lankford and S.J. Hudak Jr, 'Relevance of the Small Crack Problem to Lifetime Prediction in Gas Turbines', Int. J. of Fatigue, Vol. 9, No. 2, 1987, pp. 87-93.

7.0 ACKNOWLEDGEMENTS

The authors gratefully acknowledge the useful discussions held with various colleagues from Chalk River Laboratory of AECL. The authors are also grateful to a number of people from Canadian industry and universities for their technical contributions and to the Office of the Chief, Research and Development of the Department of National Defence for financial support of gas turbine engine structural integrity programmes at NAE.

IMPROVEMENTS IN THE PERFORMANCE OF A REGENERATIVE GAS TURBINE WITH BLADE REJUVENATION AND MODIFICATIONS TO THE COOLING AND SEALING SYSTEMS

BY

D. LITTLE
LIBURDI ENGINEERING LIMITED
HAMILTON, ONTARIO.

ABSTRACT

TCPL installed 9 Westinghouse W62RM gas turbines in their pumping system in the early 1960's. By 1980, with the price of fuel escalating and the performance of the units deteriorating with age, TCPL was seriously considering replacement with modern higher efficiency units.

When a January 1984 performance test on one unit showed power 33% and overall thermal efficiency 24% below rated (well below W62RM fleet average), Liburdi Engineering was asked to assess the unit and data and determine what, if anything, could be done to restore performance.

Over a two-year period, a series of modifications including compressor reblading, turbine blade rejuvenation, and improvements to the sealing and cooling systems were implemented and tested on two units. The latest test in November of 1986, with all modifications installed, showed the power to be 4% greater than, and the overall thermal efficiency equal to, rated. During the November test, two additional modifications were identified which could, when implemented, give a further 11% increase in power and 5% increase in efficiency.

This report describes the analysis, explains the contributions to performance improvement, and summarizes engine test data.

INTRODUCTION

The January 1984 test results which prompted the W62RM performance improvement effort are summarized in Figure 1. The TCPL test data, when referred to I.S.O. ambient conditions, indicated that only 3.6mw was being produced by the unit at its turbine inlet temperature limit of 788°C. This was 33% less than the original 5.4mw rating, and was being produced at 21.7% overall thermal efficiency instead of the rated 28.56% efficiency. The unit had deteriorated significantly from its last full performance test in 1976 and had reached the lowest level of performance yet tested for W62RM's in the fleet.

A major overhaul was scheduled for the spring of 1984 but past TCPL experience had shown that the overhauls did not usually result in significant performance gains; - they were mainly for mechanical reasons, such as bearing de-coking.

Liburdi Engineering was retained at this time to assist TCPL in deciding what action could be taken during the upcoming overhaul, and to determine what performance improvements could be expected.

GAS TURBINE COMPUTER MODEL

The W62RM is a two-shaft, regenerative cycle gas turbine as illustrated in Figure 2. It was designed in the late 1950's and consists of an eleven-stage axial compressor designed for a 4:1 pressure ratio, a two-stage compressor turbine and a single-stage power turbine. The rated thermodynamic turbine inlet temperature of 788°C gives an exhaust temperature of 511°C.

As shown in Figure 3, a computer model of the unit was set up linking all the major components of the gas turbine with the primary gas path system and including all identifiable cooling, sealing, and leakage air paths.

The axial compressor characteristic was defined as shown in Figure 4, where flow, pressure ratio, speed and efficiency are all referenced to design point (at constant inlet conditions). The actual characteristic shown here is the result of considerable refinement, using TCPL test data, of an initially assumed map.

The turbine characteristics were determined from gas path dimensions and actual blade shapes using a meanline aerodynamic analysis program developed by Liburdi Engineering Ltd. Typical characteristics are shown in Figure 5, where swallowing capacity and efficiency are related to pressure ratio, inlet temperature and speed.

PRELIMINARY ANALYSIS

The performance program was run and component design point values adjusted until the quoted design point rating of the W62RM was duplicated. Then, to determine what the expected unit 86A2 performance should be, the program was run at I.S.O. conditions using a 10% open power turbine characteristic, instead of the standard 5% open characteristic, and the actual load-speed curve of the power turbine-driven compressor. The results, plotted in Figure 6, show that the unit was actually 38% low in power and 23% low in efficiency.

To try to determine what was causing this very large shortfall in power, internal flows and component performances were varied until the measured test data was reasonably duplicated. The data could be duplicated if:

A. the axial compressor flow and efficiency were both degraded 4% to 8% from the expected new, clean values,

B. an extra (above normal expected cooling flows) 15% to 20% of compressor inlet flow bypassed the regenerator.

Restoring compressor performance by cleaning and/or replacing compressor blading would give a large increase in power, so TCPL planned to replace the rotor with a refurbished spare, if necessary, during the overhaul. However, the excessively large regenerator bypass flow posed a more difficult problem in that the actual leakage paths had first to be identified.

GAS TURBINE OVERHAUL

When unit 86A2 was opened during the overhaul, the axial compressor blading was found to be moderately fouled, as evidenced by the blackish deposits on the IGV's shown in Figure 8. In addition, row 1 and 2 blades were moderately eroded along their leading edges, and row 1 to 3 vanes had significant leading edge corrosion/erosion along leading edges (Figure 8). The combined effect of the wear and fouling could easily have lead to the degradation in flow and efficiency that had been suspected. Therefore, the replacement rotor with its new blades was installed in the unit.

Examination of the combustor shell region for possible regenerator bypass leakage paths was then undertaken. Figure 9, upper photo, shows a broken pantleg gasket which would have allowed compressor delivery air from the right-hand duct to leak into the regenerator return air duct on the left-hand side.

Opposite the pantleg duct is a blanking plate shown in the lower left photo of Figure 9. (The W62RM was built with symmetrical side openings for either left or right-hand regenerator piping). From the colour pattern, it was obvious that the plate was warping during operation and allowing leakage across the combustor dividing wall.

Each transition is held within a support ring bolted to the combustor shell dividing wall (dividing compressor delivery air from regenerator return air), but there is no positive seal provided between the outside surface of the support rings and the compressor diffuser outer shroud. Thus, flow leaked again as shown in the lower right photo in Figure 9.

Field modifications involving reinforcement, clamping and packing were made during the shutdown to stop all the leakages evident in Figure 9.

The transitions (flared pieces connecting the exit of the combustor basket to the compressor turbine inlet) had leakage paths at both ends. On the upstream end, the single thickness, flat leaf, spring seals contained free area for compressor delivery air to leak across the dividing wall as shown in Figure 10. The top photo in Figure 10 shows a darkened area on the outer shroud of the Row 1 diaphragm indicating considerable leakage of compressor delivery air into the gas path. The lower right photo of Figure 10 shows a gap between the outer flange of the transition and the upstream face of the first blade ring. Nothing could be done to remedy these problem areas during the shutdown period.

In summary, the total identifiable area for regenerator bypass flow came close to that deduced in the preliminary analysis.

The turbines were fine except for a small region of tip rubbing on the row 1 blades, and removal of about 1/16" from the complete tip of the row 2 blades (Figure 11). Therefore, refurbished (tip welded, re-contoured, HIP'd) row 2 blades were assembled in the spare rotor.

JUNE 1984 TEST RESULTS

When the unit was re-assembled and tested in June 1984, a 50% increase in power was demonstrated along with a 20% increase in overall thermal efficiency. Figure 12 shows both the January '84 and the June'84 test results.

Analysis of the data indicated the following changes had occurred since January:

A. Axial compressor flow = + 9%
   Axial Compressor efficiency = + 6%
B. Compressor turbine efficiency = + 1.5%
C. Regenerator bypass flow = - 7% of inlet flow

The relative contributions to the overall performance gain of each of the improvements are also shown in Figure 12. The new compressor blading gave the largest power increase but did not improve efficiency at a constant power. The gain in compressor turbine efficiency resulting from re-tipping the blades contributed significantly to both increased power and efficiency. And the change due to reduced leakage gave the largest gain in efficiency because more of the turbine exhaust heat was being recovered in the regenerator.

Even with the excellent recovery of lost power due to the actions noted, the analysis showed that there remained approximately 13% of inlet air bypassing the regenerator, so the potential definitely remained to achieve further gains.

TRANSITION SEALING

To rectify the leakage areas at both ends of the transitions, a sealing package was designed as sketched in Figure 13. At the upstream end, double overlapping, spherically-shaped spring seals were designed. At the exit, the outer flange gap shown in Figure 10 was sealed by a spring-loaded angle seal held in contact with the transition with spring clips and against the blade ring by coil springs. Between transitions, flat spring side seals were developed.

A spare set of transitions was modified by Liburdi Engineering and returned to TCPL for field testing.

To evaluate the performance gain resulting from the transition modification, TCPL conducted another instrumented performance test of unit 86A2 in December '84, just prior to bringing the unit down for transition replacement, and then again in January '85, immediately after replacement.

As can be seen in Figure 14, the December '84 results, when referred to I.S.O. conditions, showed that the unit had dropped 1.15mw in capability in six months of operation. The January '85 tests did show a sizeable performance gain over December, but the overall degradation was disappointing. Preliminary analysis pointed to the fact that a loss in axial compressor flow and efficiency appeared to be the main culprit, so another test was conducted in April'85 when there was no possibility of icing.

Extensive analysis of the four sets of test data summarized in Figure 14 was then conducted to separate the transition sealing modification effects from all others. All the test points could be reasonably modelled with the values tabulated in Table II.

Compressor fouling was the primary cause of performance loss showing that even in the clean Northern Ontario atmosphere, severe penalties are suffered as the result of airborne contaminants. The total regenerator bypass flow was reduced by about 4.5% of inlet flow due to the regenerator sealing changes. Compressor turbine efficiency dropped 1.3 points sometime in the first six months of operation and then held constant. It was surmised that the chronic row 2 blade tip rubbing had occurred soon after rebuild.

Separating transition sealing effects from all other variables gave a 10% power and 6% efficiency improvement due to transition seals alone. Extrapolating the June'84 test results an extra 10% and 6% indicated that the unit could achieve design power at close to design efficiency.

ADDITIONAL PERFORMANCE IMPROVEMENTS

The recurrent row 2 blade tip rubbing problem was studied and in combination with other experience on similar turbines, it was decided that casing distortion due to gas path leakage into the blade ring cavities could be the culprit. A blade ring cavity ventilation system such as that of Figure 15 was suggested.

It was suspected that the row 1 diaphragm sealing configuration shown in part A of Figure 16 allowed hot gas to flow from upstream high pressure to downstream low pressure under the inner shroud, thus reducing stage 1 efficiency. In addition, hot gas flowing over the torque tube exit labyrinth seal carrier would tend to heat up the carrier and open up the clearance under the seal, allowing excessive compressor delivery air to bypass the regenerator. A static seal modification was designed as shown in Figure 16, part B to seal the inner shroud.

Measurements and analysis on similar W52/W62/W72 gas turbines had pointed to a possible row 2 disc cooling problem. A disc cooling modification had been designed (Figure 17) to take compressor delivery air from the combustor shell, externally cool it, and then direct it into the row 2 disc upstream cavity through an improved row 2 diaphragm seal.

These modifications were fabricated and supplied to TCPL to test on another unit in combination with all previous modifications.

EVALUATION OF COMPLETE MODIFICATION PACKAGE

A second unit, 62A1, was chosen to test all of the modifications previously outlined. It incorporated the standard 5% open power turbine nozzle; therefore, using the actual power turbine load-speed characteristic, the expected performance was calculated and is shown in Figure 18.

The turbine was baseline tested in May '86 before modification and was found to be 9% low in power and 10% low in efficiency. Analysis of the data indicated that, as shown in Figure 19;

A - the compressor performance exceeded design expectations,

B - there was approximately 13% of inlet air not reaching the regenerator,

C - the compressor turbine efficiency was 1.5% low.

Tests on the unit in November '86 demonstrated the performance gains shown in Figure 20. Power was brought up to design expectations, with efficiency falling just short of the 28.3% target.

Instrumentation which had been included in the modified components showed the following:

1. The blade ring cavity ventilation system achieved an average drop in blade ring cavity temperature of 84°C for cavity #1 and 146°C for cavity #2.

2. The row 1 diaphragm static seal stopped all leakage under the inner shroud.

3. The combined effects of cooler blade ring cavity and improved row 2 sealing reduced the leakage of hot gas under the diaphragm so much that cooling air was not needed.

Figure 21 shows the final deduced compressor turbine cooling and leakage flows for this November test expressed as percentages of inlet air flow. The total regenerator bypass flow was reduced to about 4% of inlet flow.

The analysis showed that the axial compressor performance did not change from May to November and thus, the total performance gain was due to reduced leakage flows and improved compressor turbine efficiency (due to the re-tipped blade and cavity ventilation).

The post-modification performance data was gathered after the occurrence of a full-load trip and hot restart and indicated that very tight hot running turbine clearances were being achieved and held. The tight clearances gave a compressor turbine efficiency about 3% greater than that before modification (with the rubbed row 2 blade).

FUTURE MODIFICATIONS

Re-examining Figure 21, it can be seen that there remain two areas of excess internal leakage, namely:

A - the total flow necessary to stop hot gas ingestion into the blade ring cavity is about 2.7% of inlet flow. This is well in excess of a "good" engine quantity and should be reducable through blade ring sealing modifications, and,

B - the total flow leaking under the torque tube exit labyrinth seal is about 2.8% of inlet flow; much more than necessary to provide adequate row 1 disc cooling. By fabricating a 2-step labyrinth seal with the nominal cold clearance reduced from 0.100" to 0.035", the leakage can be reduced to 1.1% of inlet airflow.

Incorporating these two improvements into the engine tested in November '86 would give the performance shown in Figure 22. The maximum I.S.O. power would exceed design by 14% with the efficiency exceeding design by 5%.

CONCLUSIONS

The W62KM gas turbine has achieved substantial performance improvements through a systematic program of test, analysis and modification. These improvements have been possible due to the cooperative effort between the operator and an independent gas turbine engineering company.

TABLE I

SUMMARY OF VALUES NEEDED TO MODEL THE TEST RESULTS

TEST	JUNE 1984	DEC. 1984	JAN. 1985	APRIL 1985
Compressor Map Design Point Flow (lb/sec)	101.8	96.9	96.4	95.0
Compressor Map Design Point Efficiency (%)	85.1	81.7	82.0	81.0
Combustor Dividing Wall Leakage as a percentage of Inlet Flow (%)	5.2	5.2	5.2	6.2
Transition Mouth Leakage as a percentage of Inlet Flow (%)	7.5	8.5	3.8	2.3
Gas Generator Turbine Design Point Efficiency (%)	87.8	86.5	86.5	86.5
Regenerator Effectiveness (%)	82.0	82.0	79.5	81.0

UNIT 86A2
THERMAL EFFICIENCY vs SHAFT POWER

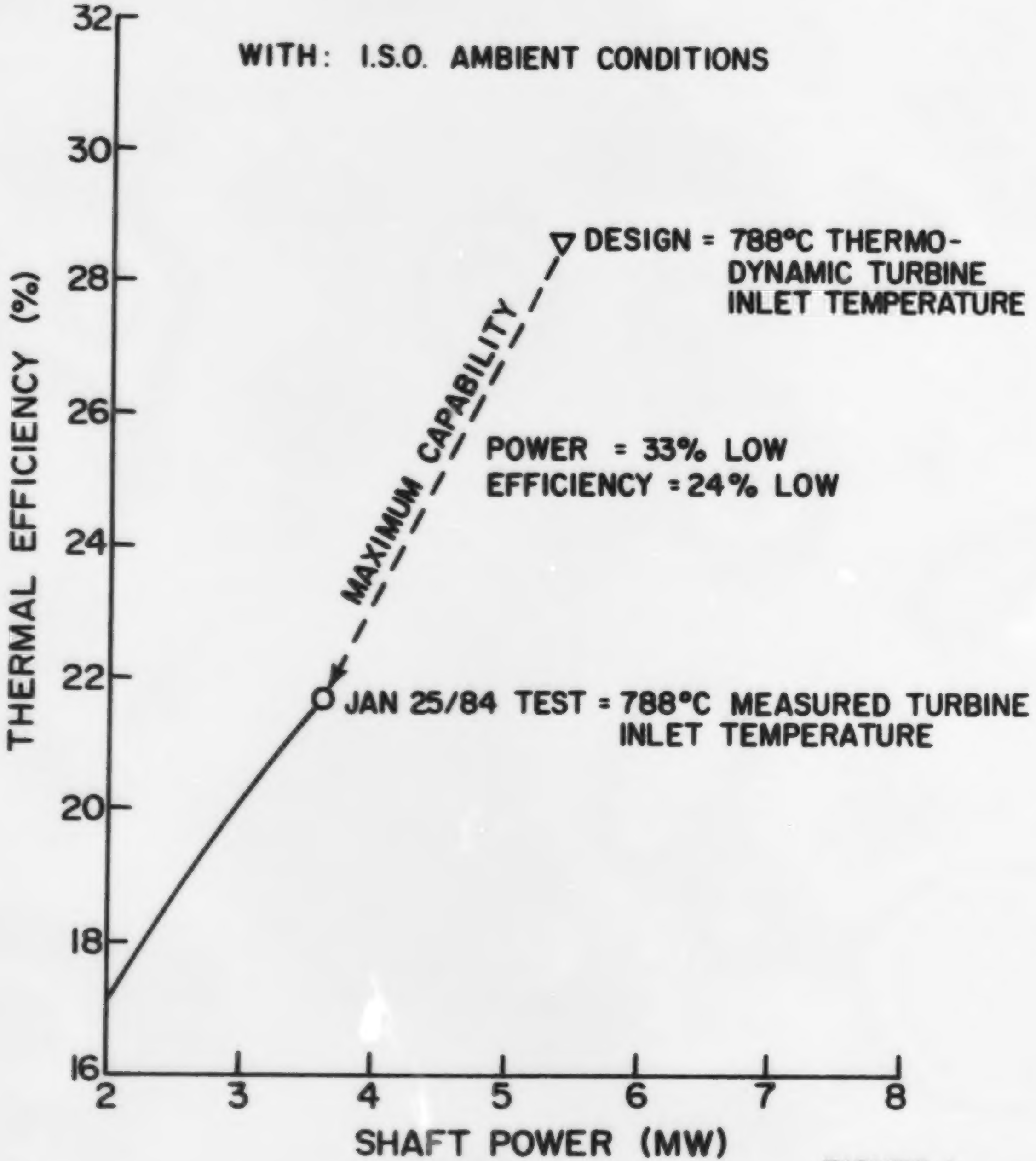
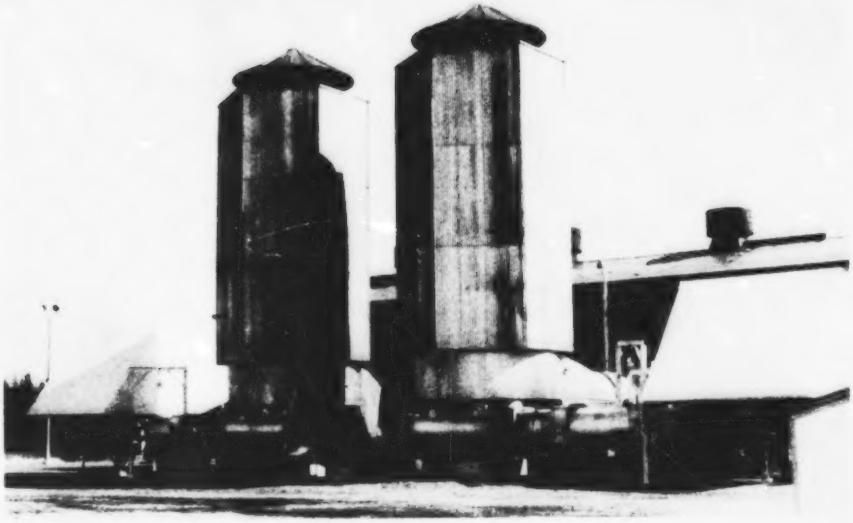


FIGURE I

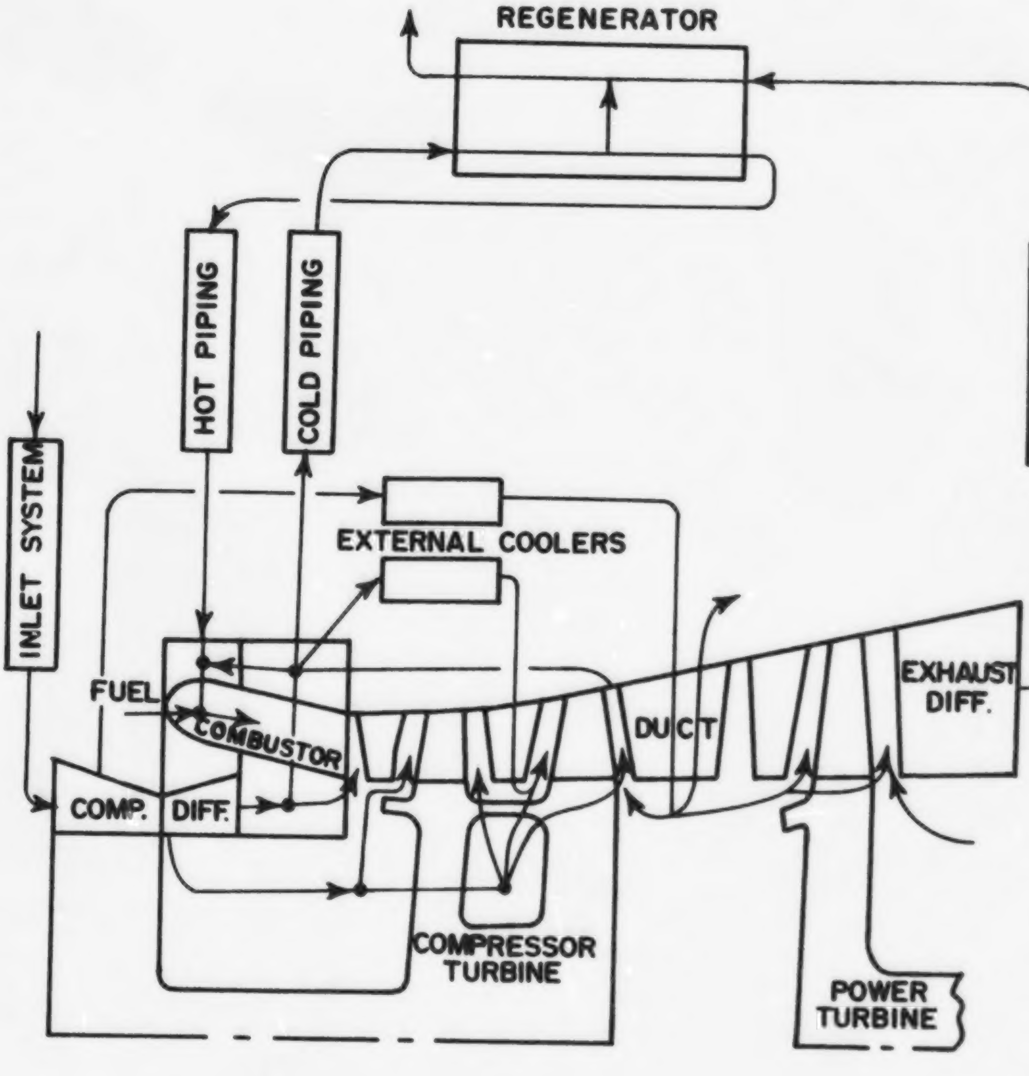
2 SHAFT - 2 STAGE COMPRESSOR TURBINE
- I STAGE POWER TURBINE



SHELL AND TUBE REGENERATORS AT TWIN UNIT STATION

W62RM INDUSTRIAL GAS TURBINE

FIGURE 2


REGENERATOR
HOT PIPING
COLD PIPING
DUCTING
INLET SYSTEM
EXTERNAL COOLERS
FUEL
COMBUSTOR
COMP.
DIFF.
DUCT
EXHAUST DIFF.
COMPRESSOR TURBINE
POWER TURBINE

PERFORMANCE PROGRAM SCHEMATIC

FIGURE 3

W62 NON-DIMENSIONAL COMPRESSOR MAP

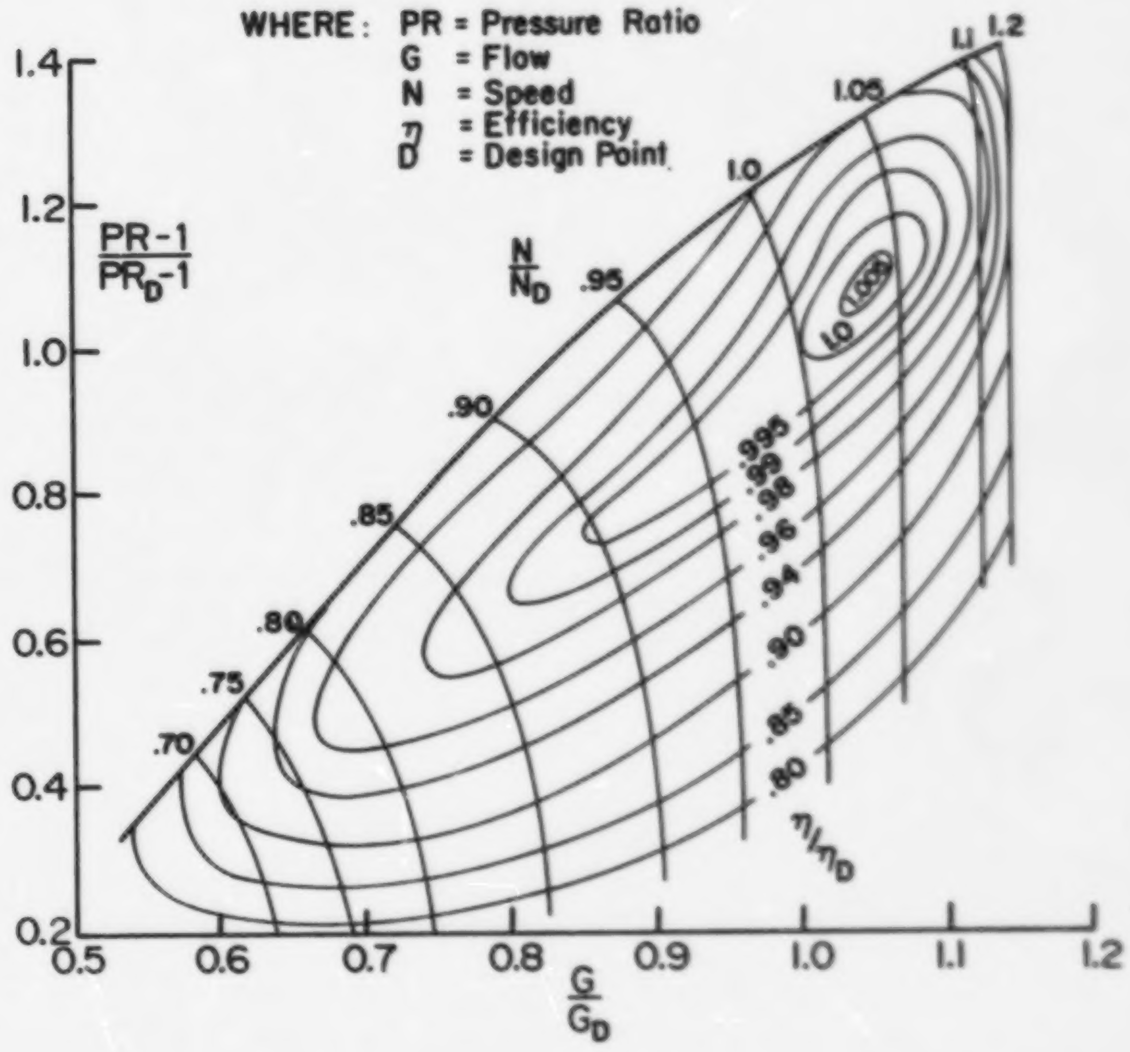


FIGURE 4

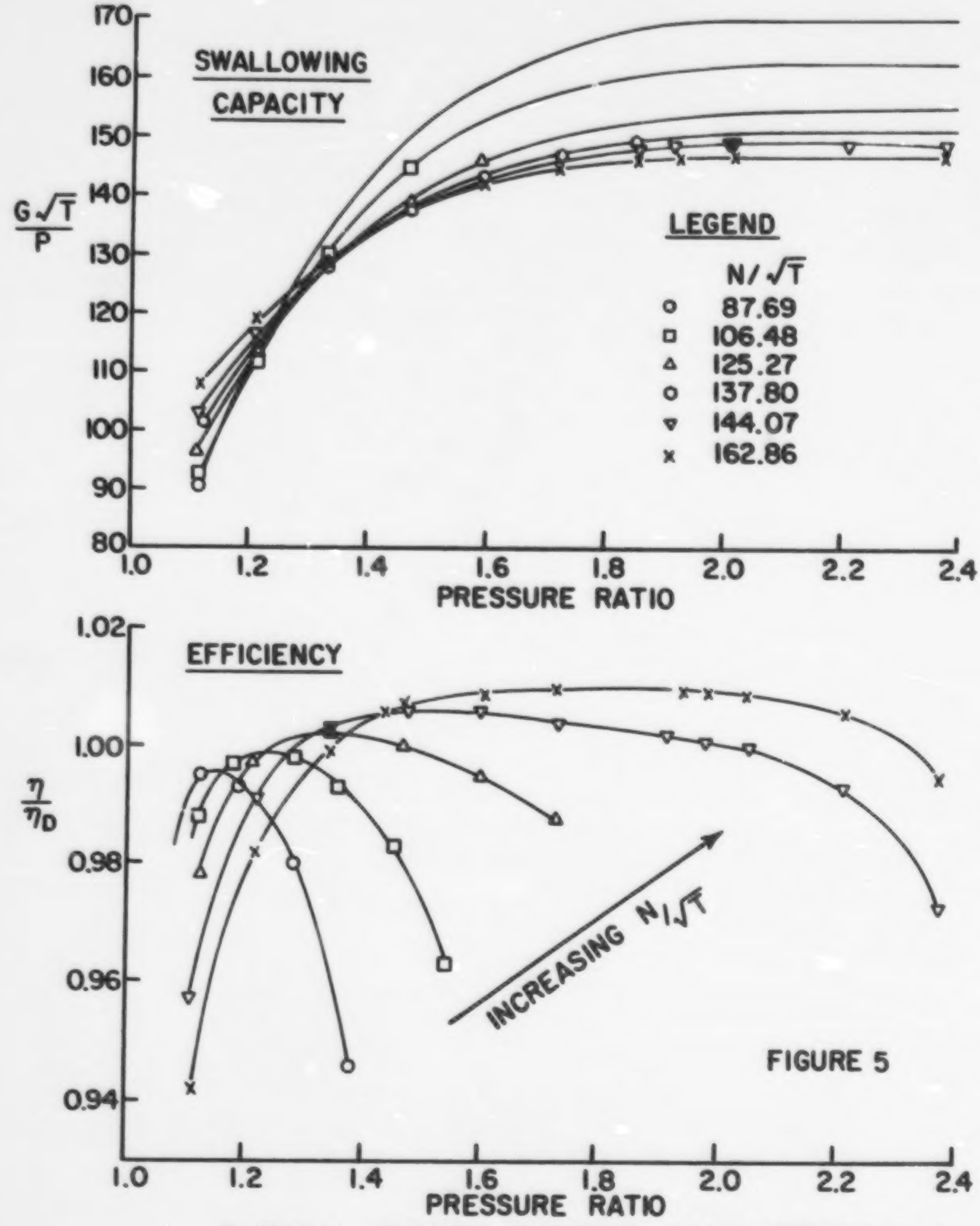


FIGURE 5

TYPICAL POWER TURBINE CHARACTERISTICS

UNIT 86A2
THERMAL EFFICIENCY vs SHAFT POWER

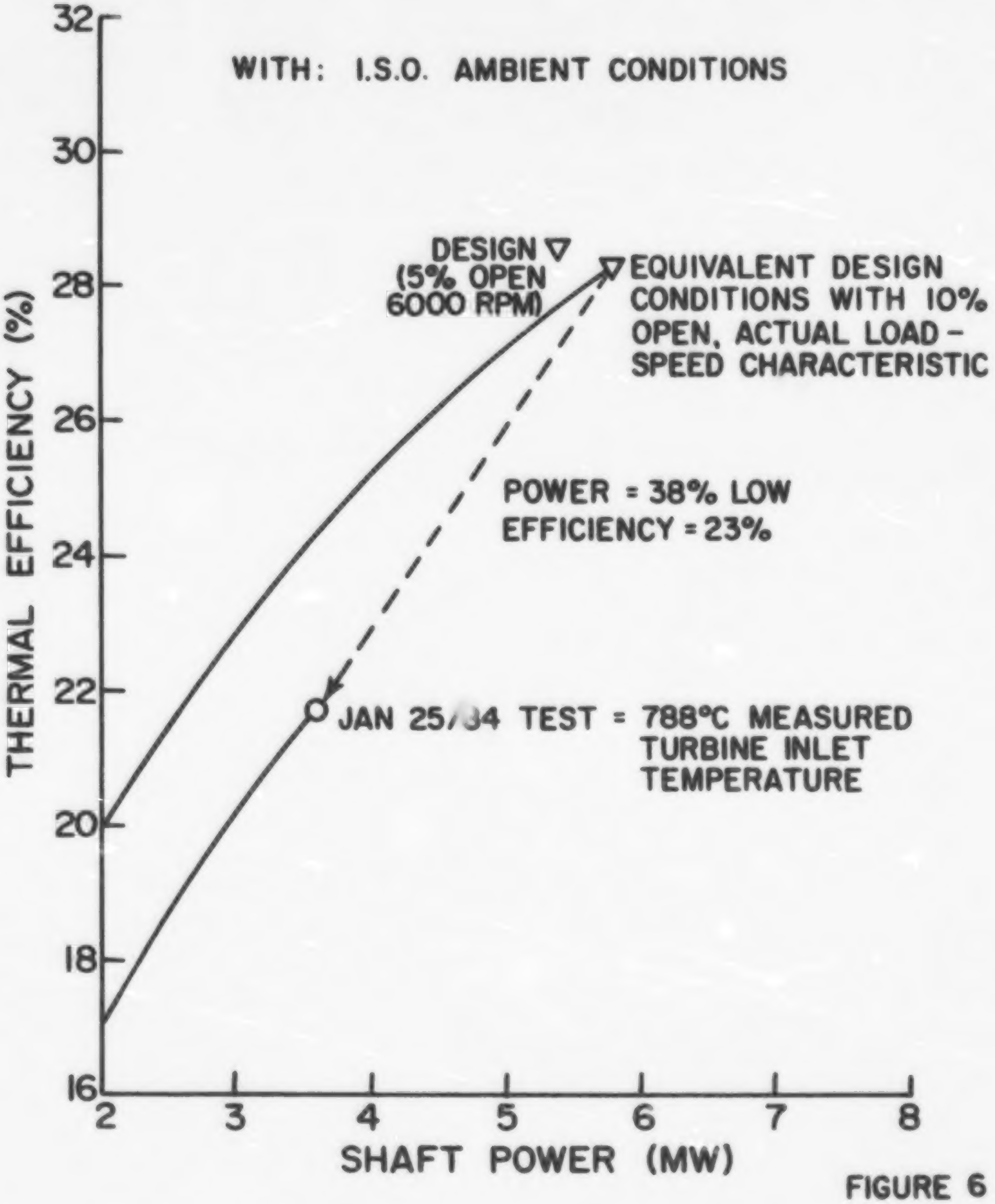
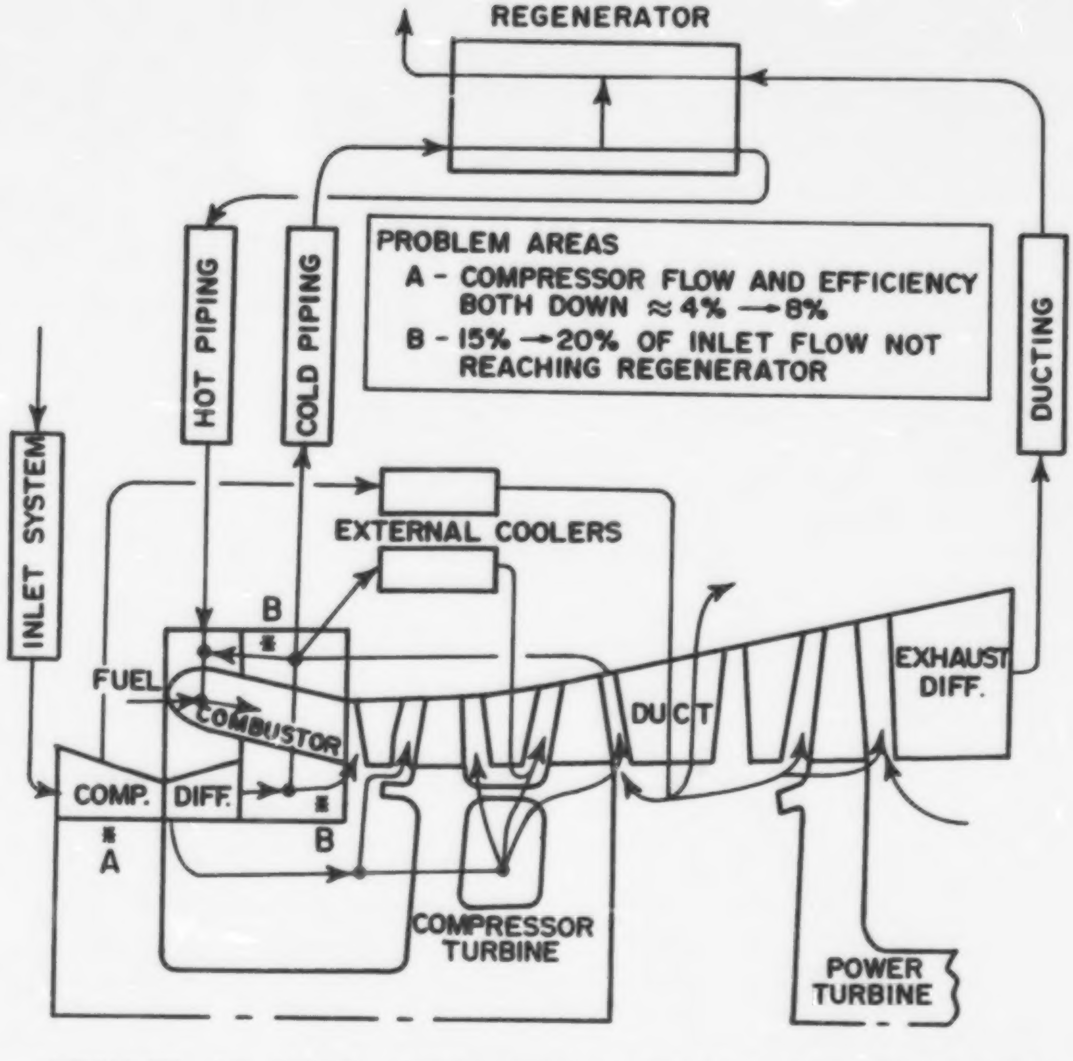


FIGURE 6


REGENERATOR
HOT PIPING
COLD PIPING
PROBLEM AREAS
A - COMPRESSOR FLOW AND EFFICIENCY BOTH DOWN ≈ 4% → 8%
B - 15% → 20% OF INLET FLOW NOT REACHING REGENERATOR
DUCTING
INLET SYSTEM
EXTERNAL COOLERS
B
FUEL
COMBUSTOR
COMP.
DIFF.
A
B
DUCT
EXHAUST DIFF.
COMPRESSOR TURBINE
POWER TURBINE

RESULTS OF FIRST ATTEMPT AT MODELLING TCPL TEST DATA

FIGURE 7

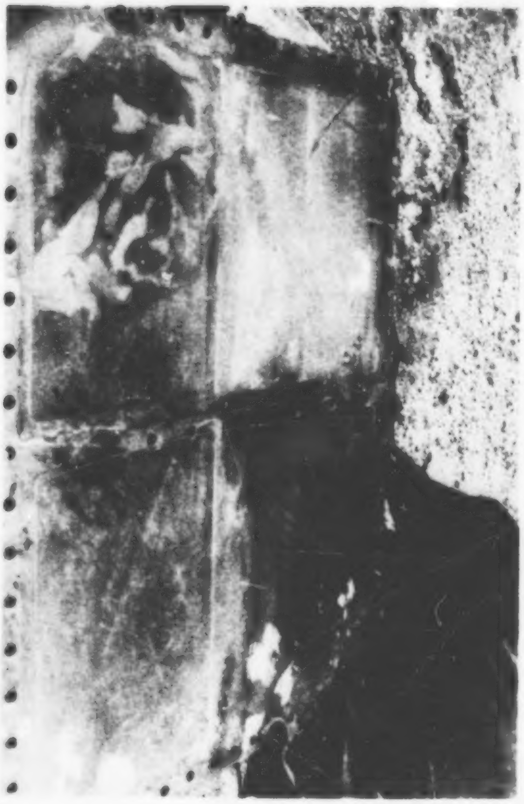
- SOME FOULING AS SHOWN ON IGV
- ROW 1 AND ROW 2 BLADES ERODED ON LEADING EDGES

ROWS 1 TO 3 VANES WITH SIGNIFICANT LEADING EDGE CORROSION

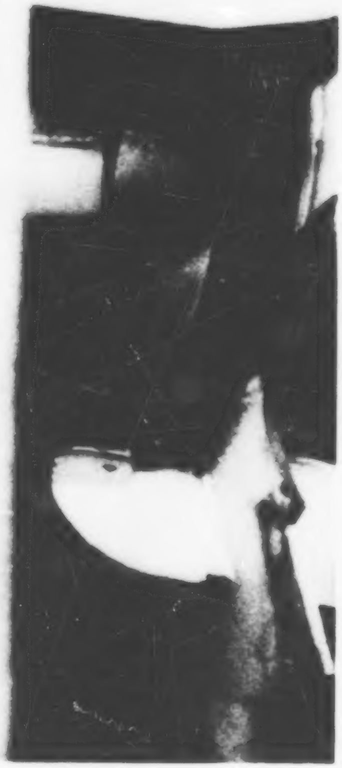
CONDITION OF COMPRESSOR

FIGURE 8

BROKEN PANTLEG GASKET



BOWING BLANKING PLATE



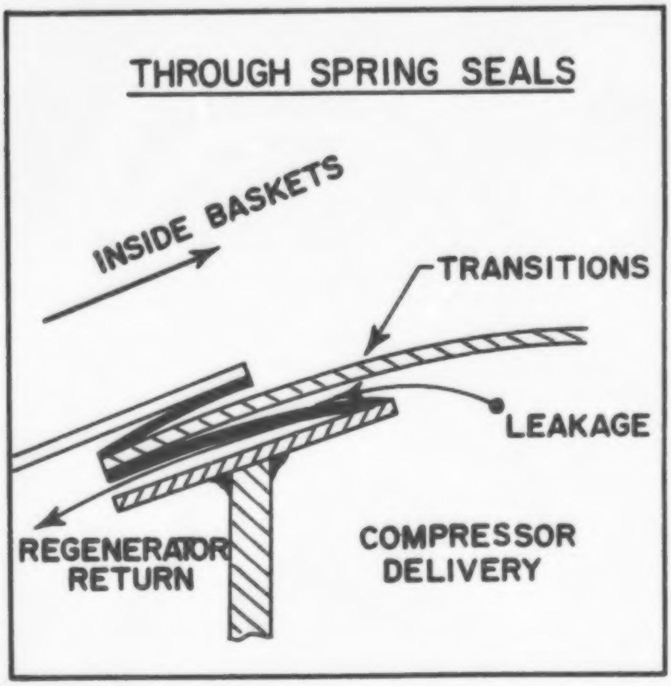
UNDER COMBUSTOR WALL
TRANSITION SUPPORT RINGS

PATHS FOR AIR TO BYPASS REGENERATOR

FIGURE 9

BETWEEN TRANSITIONS AT DOWNSTREAM ENDS

THROUGH TRANSITION OUTER FLANGE/ BLADE RING CAP

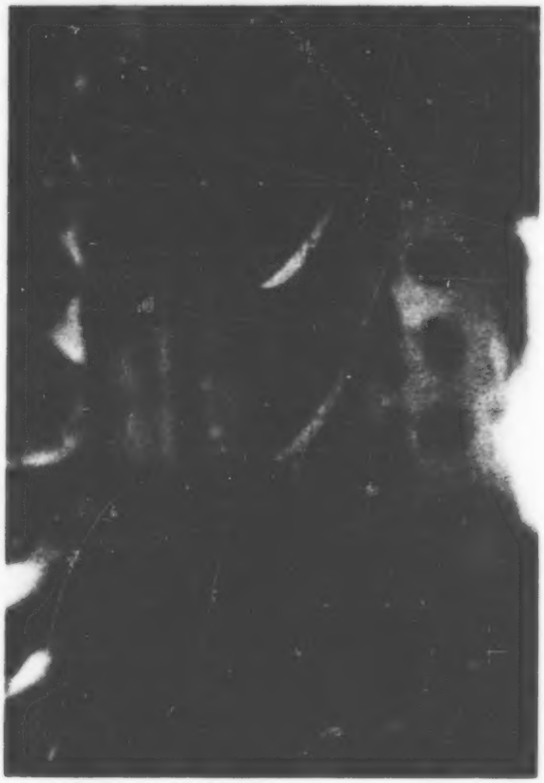


THROUGH SPRING SEALS

INSIDE BASKETS
TRANSITIONS
LEAKAGE
REGENERATOR RETURN
COMPRESSOR DELIVERY

TRANSITIONS LEAKAGE PATHS FIGURE 10

ROW I BLADES RUBBED ≈ .060" AT TRAILING EDGE ONLY

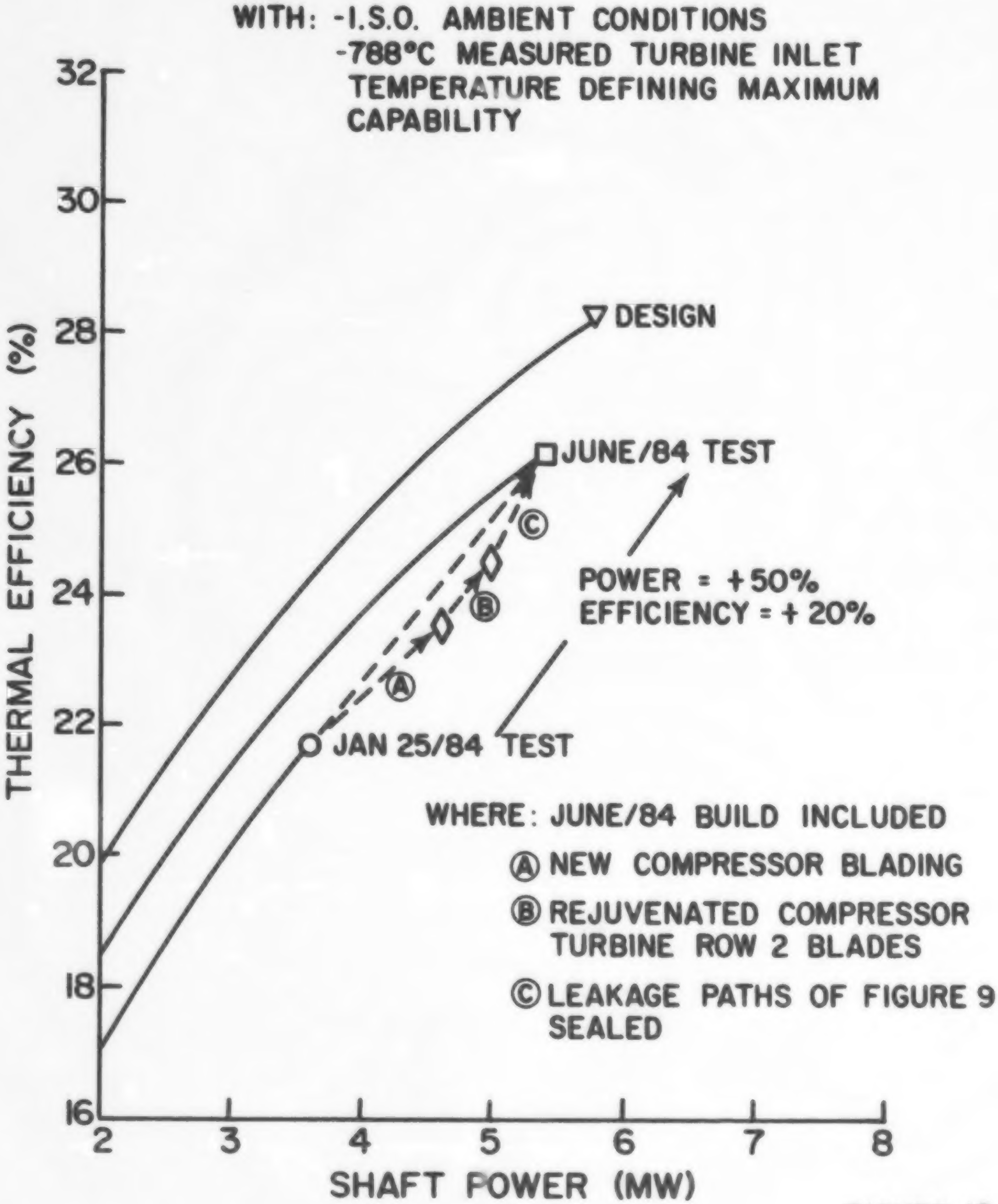


ROW 2 BLADES RUBBED≈.060'
OVER ENTIRE TIP

COMPRESSOR TURBINE BLADE TIP RUBS

FIGURE II

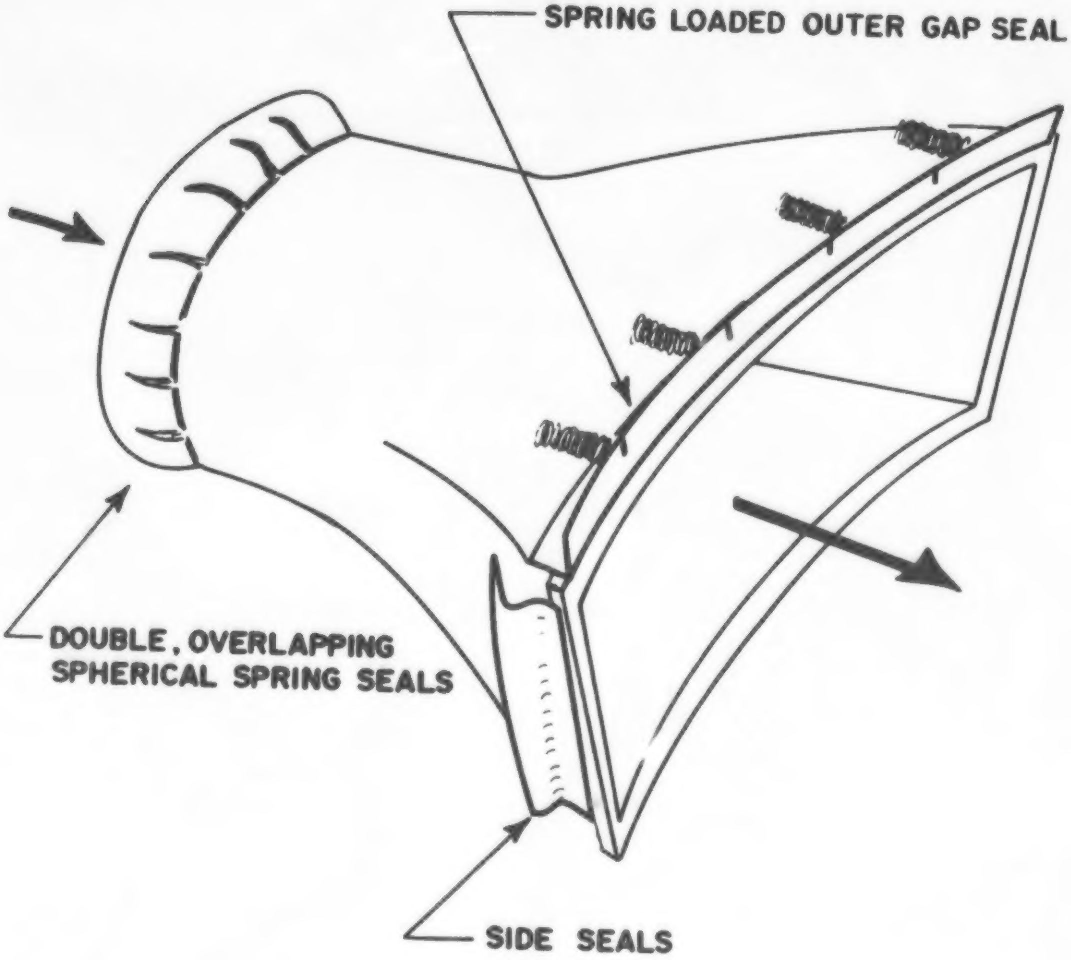
UNIT 86A2
THERMAL EFFICIENCY vs SHAFT POWER



WITH: -I.S.O. AMBIENT CONDITIONS
-788°C MEASURED TURBINE INLET TEMPERATURE DEFINING MAXIMUM CAPABILITY

WHERE: JUNE/84 BUILD INCLUDED
Ⓐ NEW COMPRESSOR BLADING
Ⓑ REJUVENATED COMPRESSOR TURBINE ROW 2 BLADES
Ⓒ LEAKAGE PATHS OF FIGURE 9 SEALED

FIGURE 12


SPRING LOADED OUTER GAP SEAL
DOUBLE, OVERLAPPING SPHERICAL SPRING SEALS
SIDE SEALS

TRANSITION SEALING MODIFICATIONS

FIGURE 13

UNIT 86A2
THERMAL EFFICIENCY vs SHAFT POWER

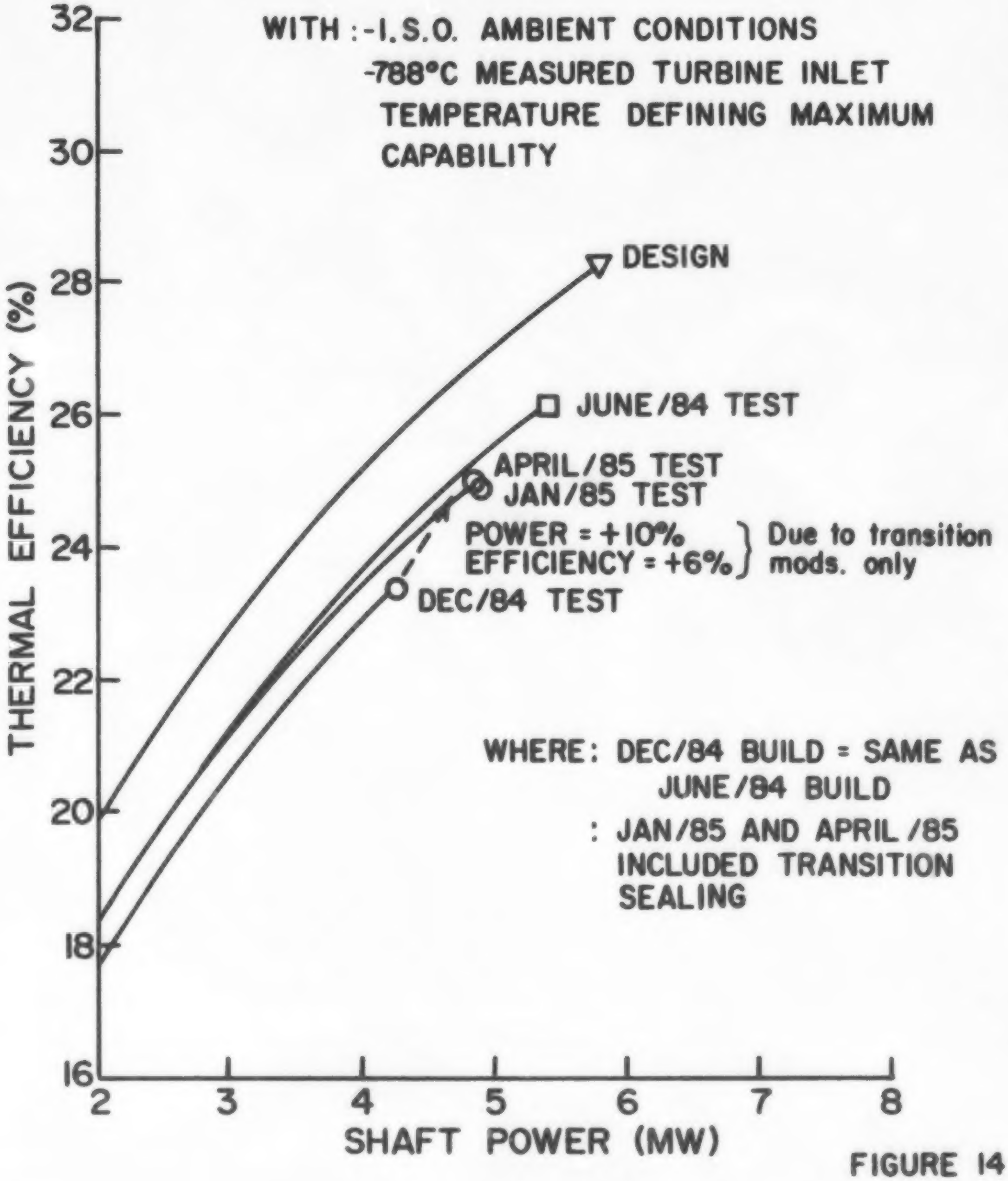
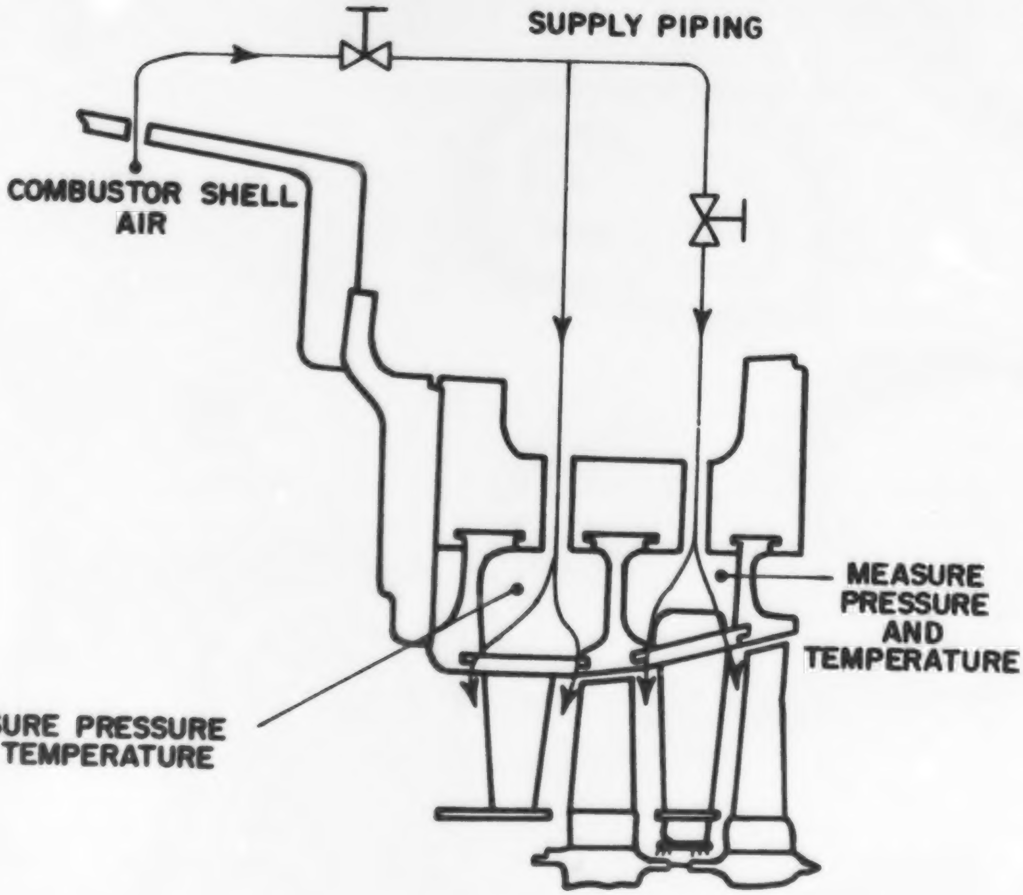
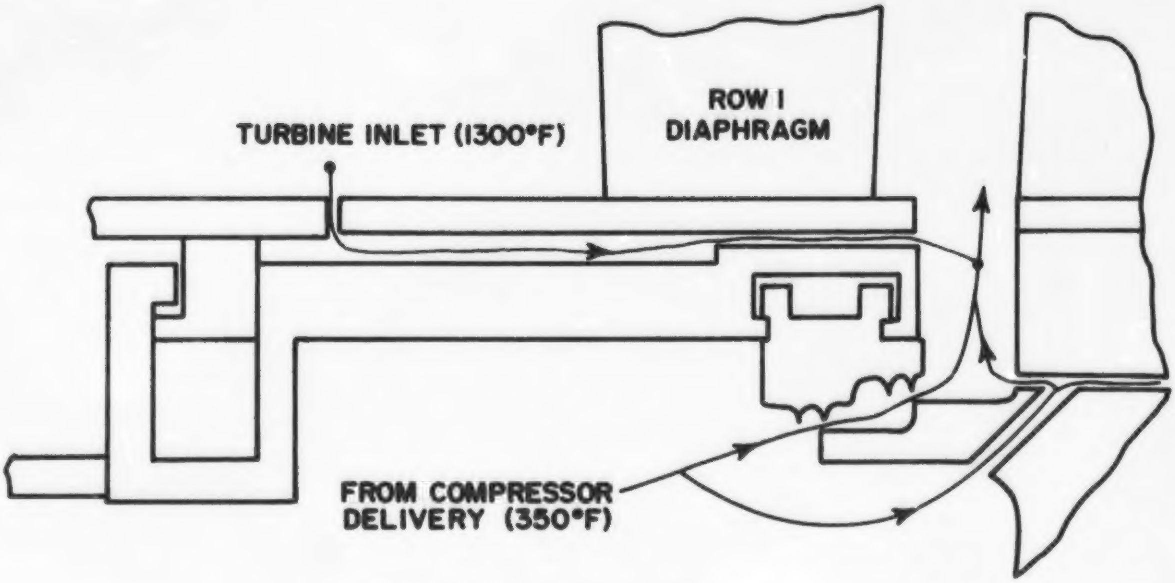


FIGURE 14

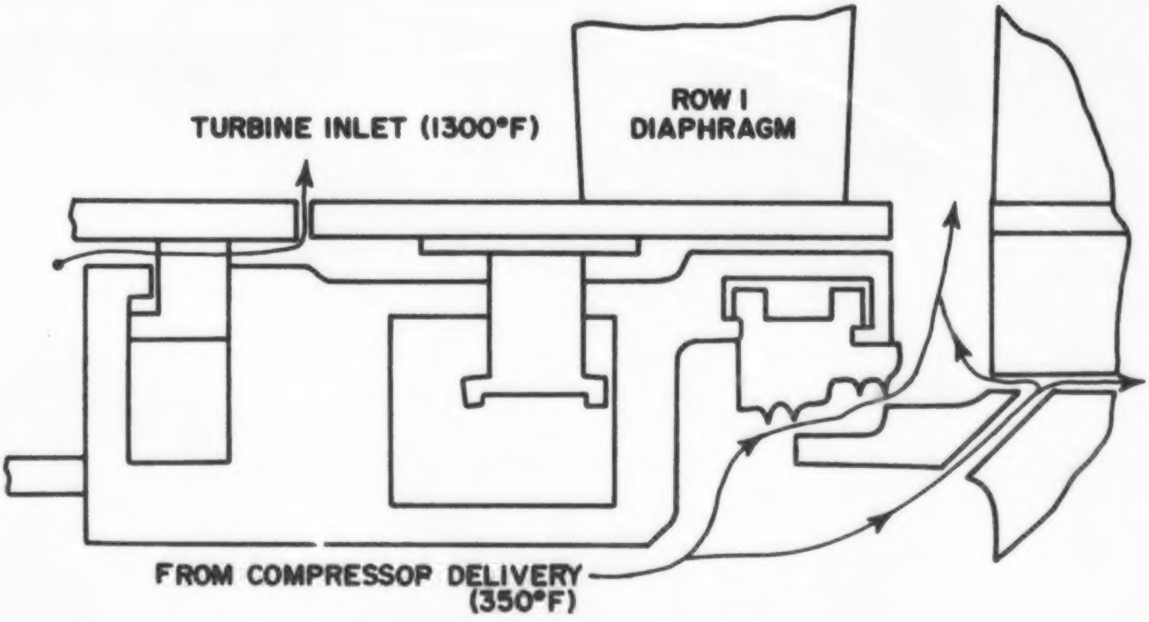

SUPPLY PIPING
COMBUSTOR SHELL AIR
MEASURE PRESSURE AND TEMPERATURE
MEASURE PRESSURE AND TEMPERATURE

BLADE RING CAVITY VENTILATION SYSTEM

FIGURE 15


ROW I
DIAPHRAGM
TURBINE INLET (1300°F)
FROM COMPRESSOR
DELIVERY (350°F)

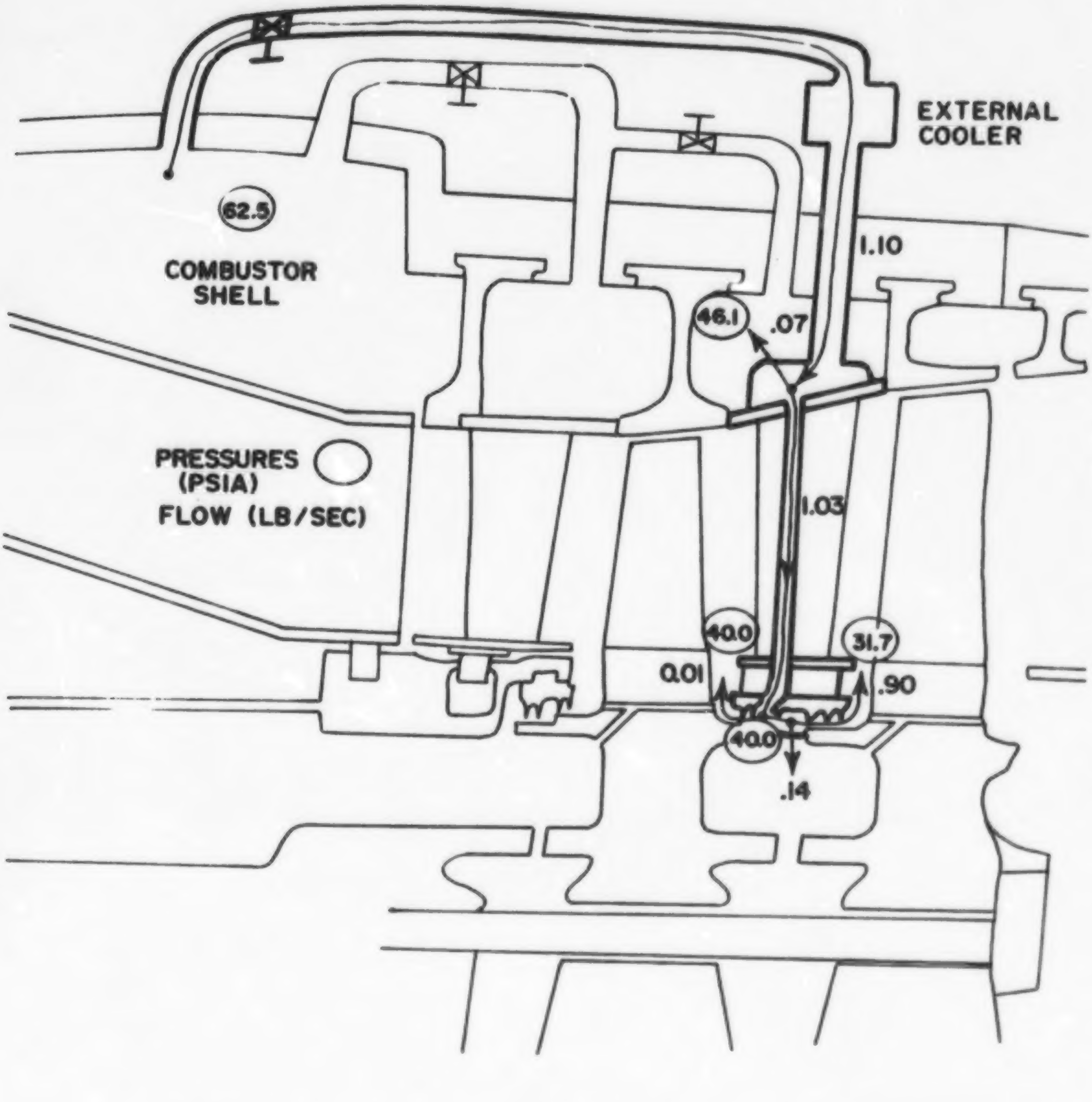
A - ORIGINAL CONFIGURATION


ROW I
DIAPHRAGM
TURBINE INLET (1300°F)
FROM COMPRESSOP DELIVERY
(350°F)

B - AFTER STATIC SEAL ADDITION

STATIC SEAL UNDER ROW I DIAPHRAGM

FIGURE 16


EXTERNAL COOLER
62.5
COMBUSTOR SHELL
1.10
46.1
.07
PRESSURES (PSIA)
FLOW (LB/SEC)
1.03
40.0
31.7
0.01
.90
40.0
.14

COMPRESSOR TURBINE ROW 2 COOLING MODIFICATION

FIGURE 17

UNIT 62A1

THERMAL EFFICIENCY vs SHAFT POWER

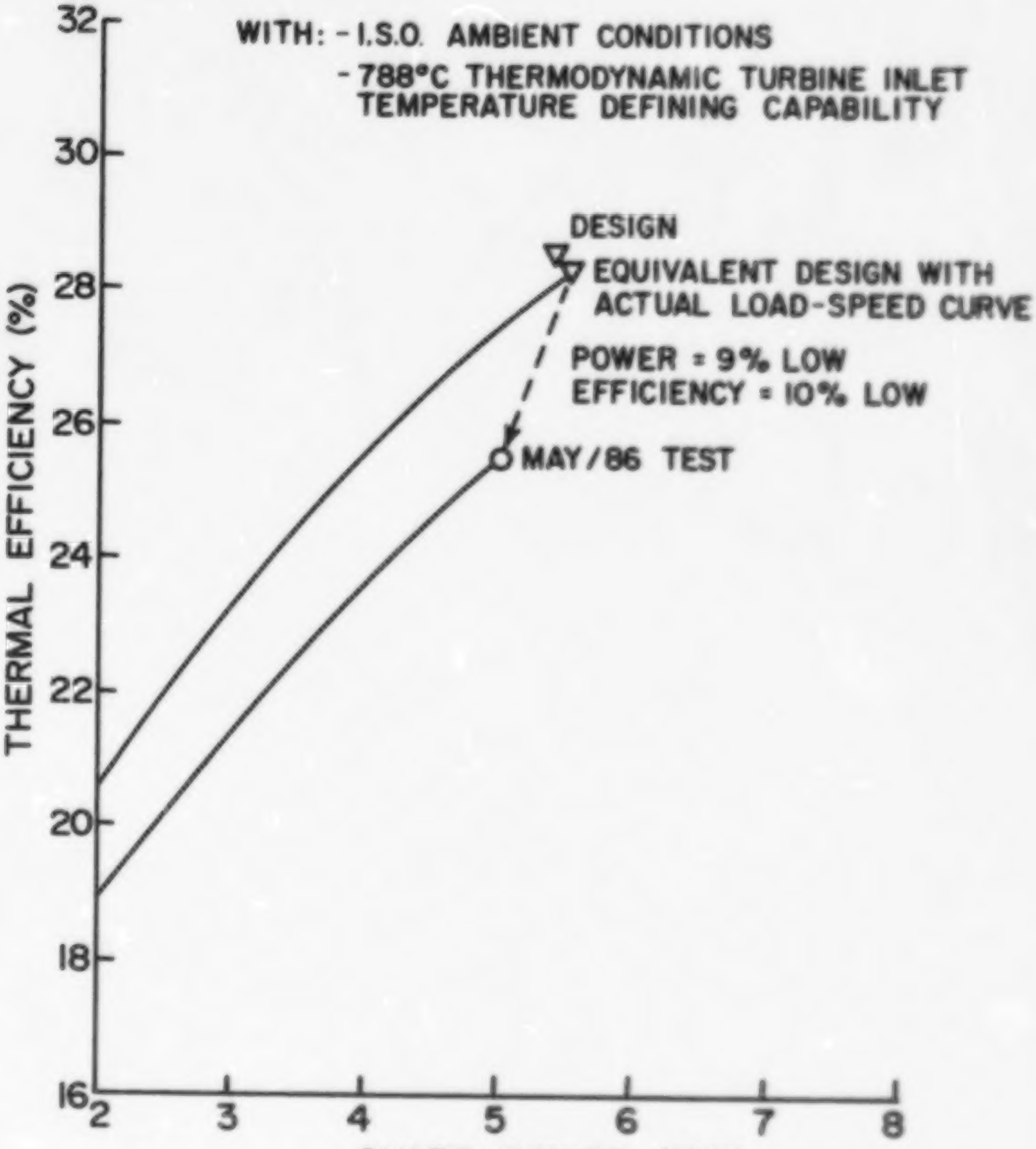
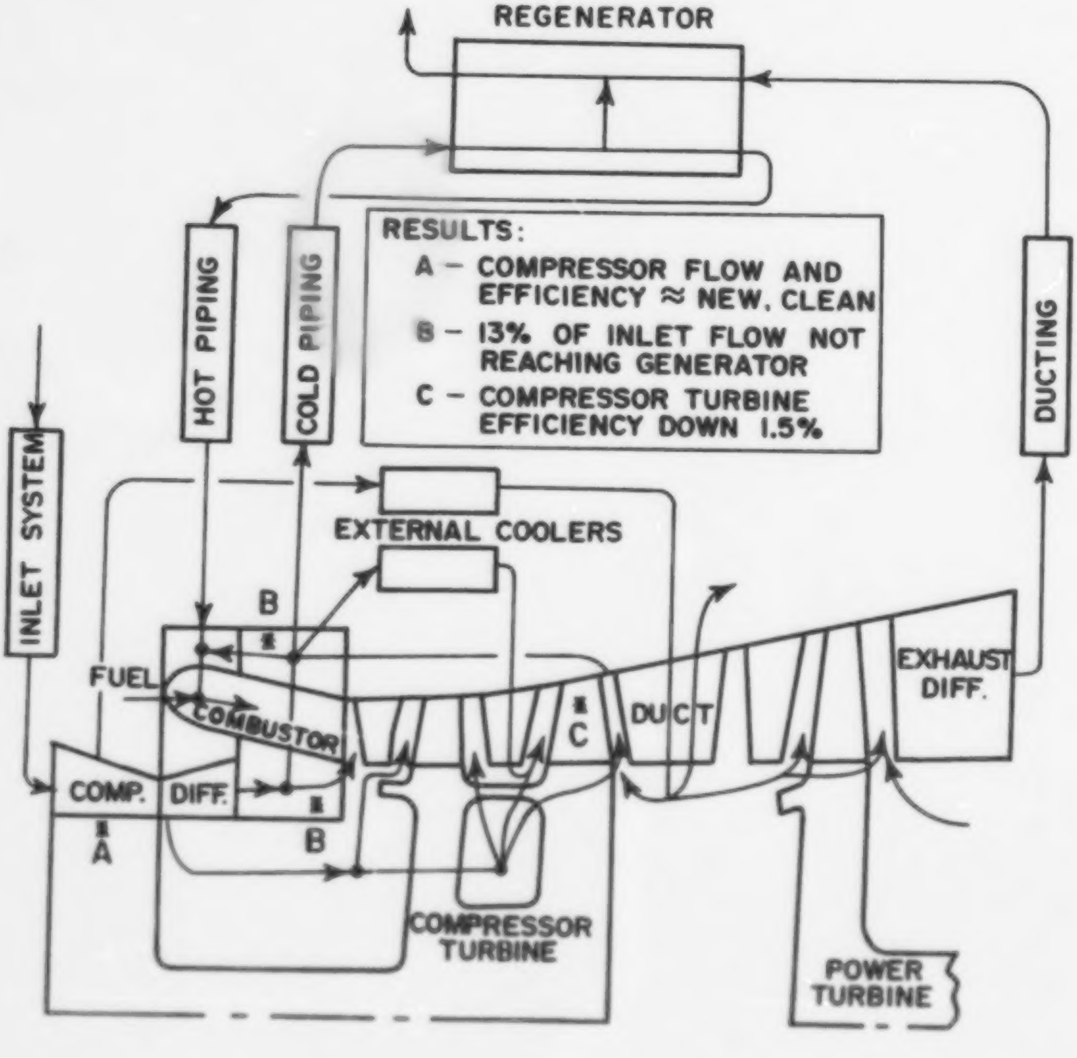


FIGURE 18


REGENERATOR
HOT PIPING
COLD PIPING
DUCTING
INLET SYSTEM
EXTERNAL COOLERS
B
FUEL
COMBUSTOR
COMP.
DIFF.
A
B
C
DUCT
EXHAUST DIFF.
COMPRESSOR TURBINE
POWER TURBINE

RESULTS:

A – COMPRESSOR FLOW AND EFFICIENCY ≈ NEW, CLEAN

B – 13% OF INLET FLOW NOT REACHING GENERATOR

C – COMPRESSOR TURBINE EFFICIENCY DOWN 1.5%

MODEL OF MAY/86 TEST DATA

FIGURE 19

UNIT 62A1
THERMAL EFFICIENCY vs SHAFT POWER

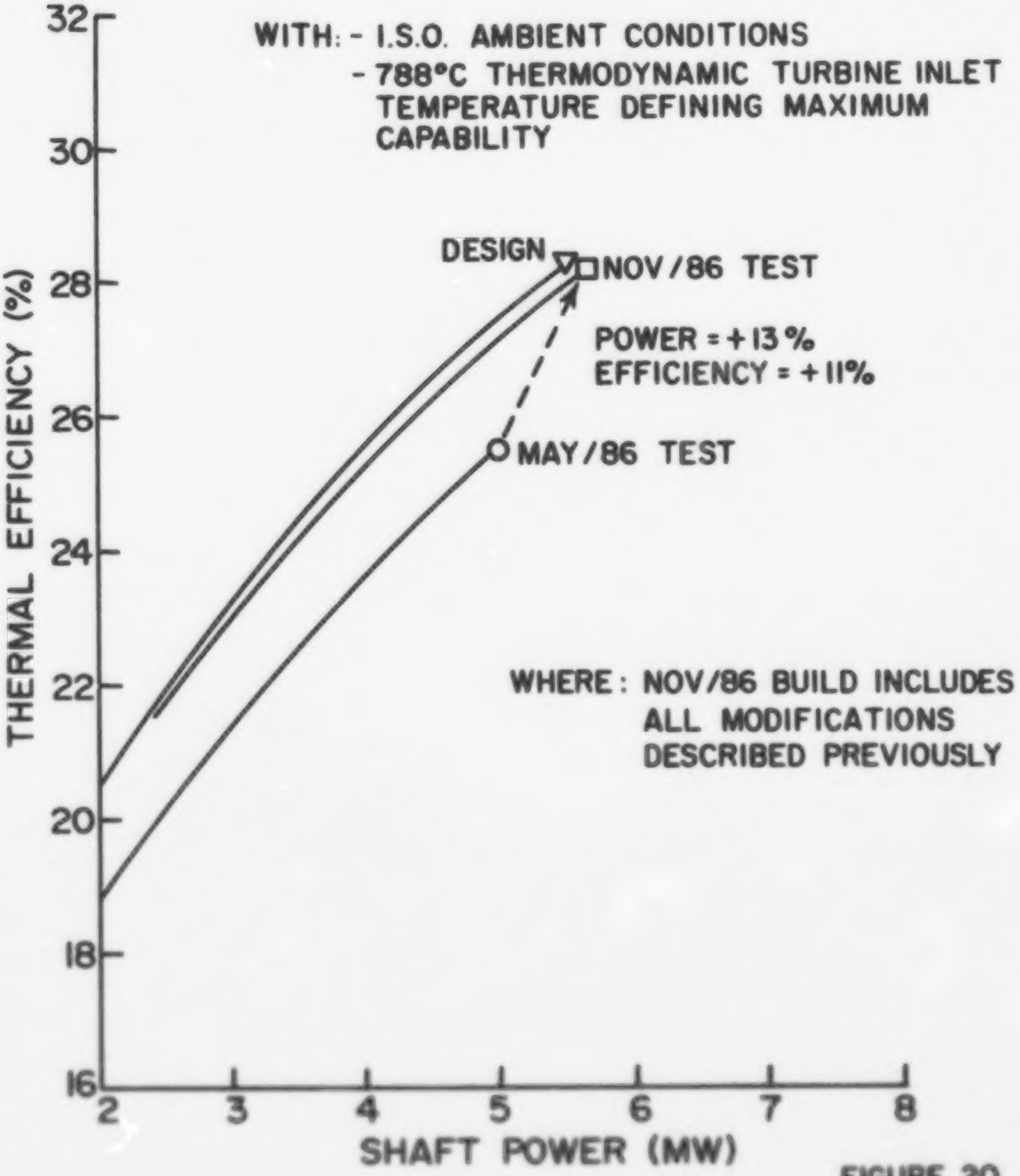
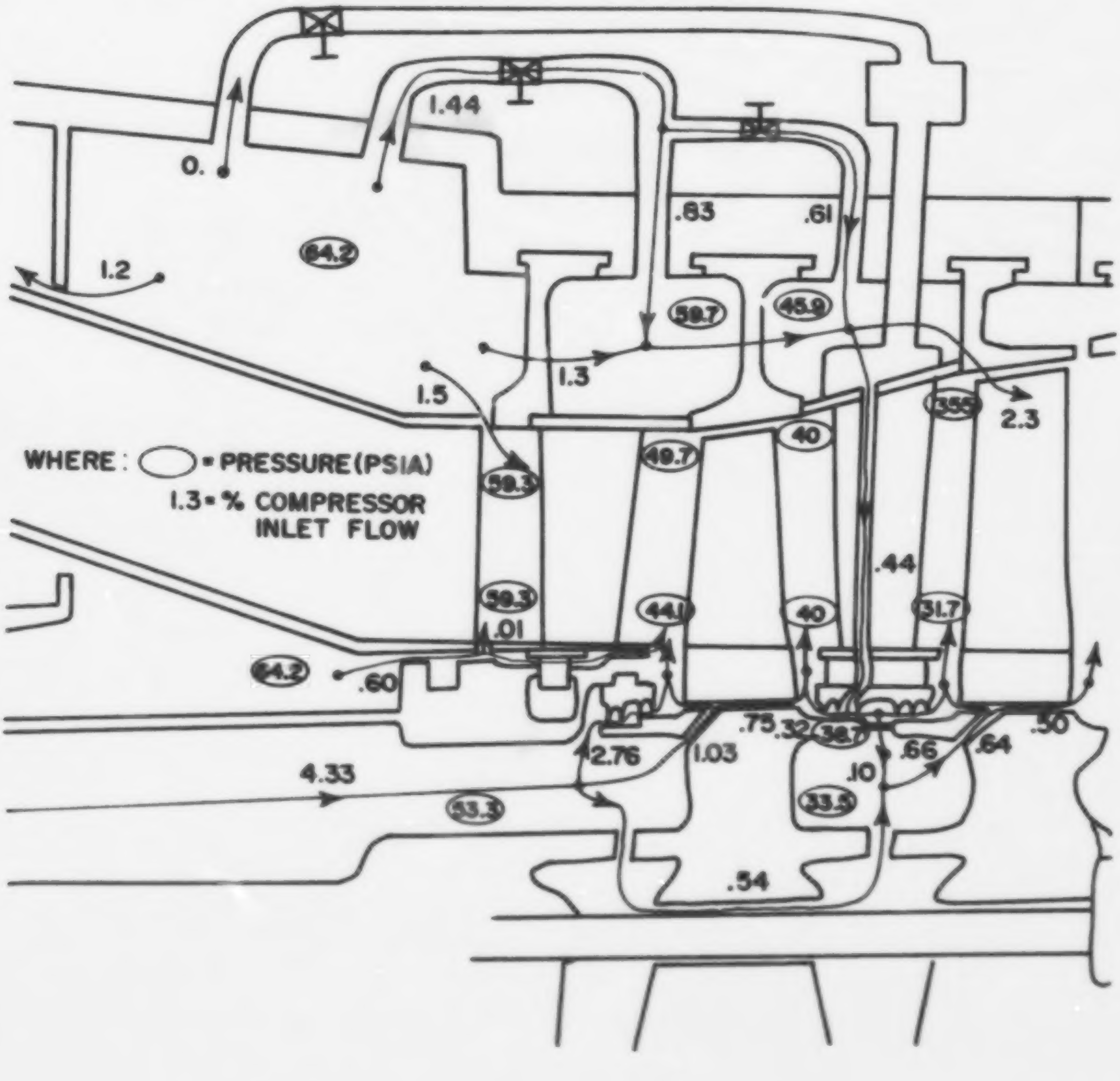


FIGURE 20

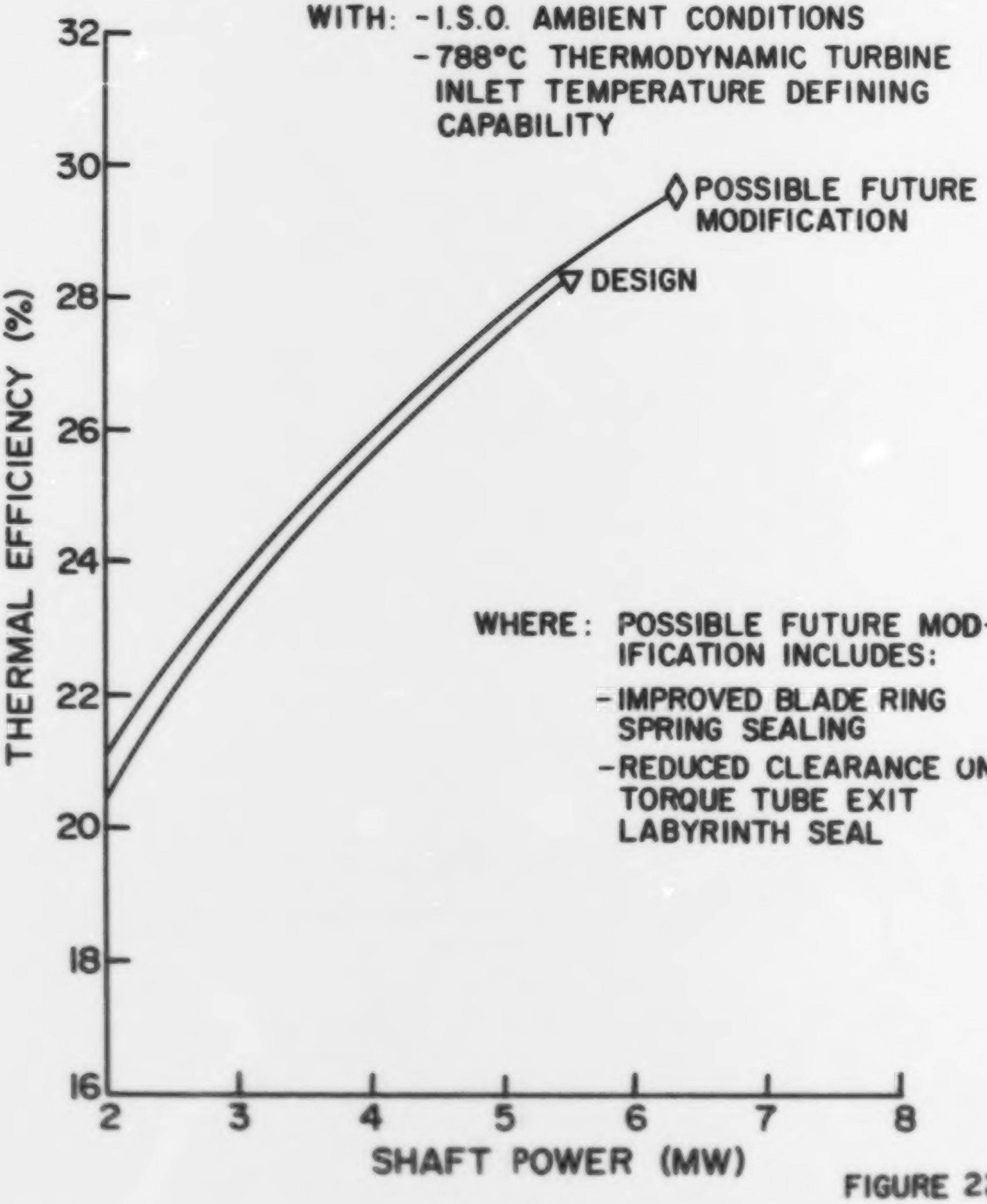

1.44
0.
.83
.61
64.2
1.2
59.7
45.9
1.3
1.5
35.5
2.3
40
WHERE : ○ = PRESSURE(PSIA)
1.3 = % COMPRESSOR INLET FLOW
49.7
59.3
.44
59.3
44.1
40
31.7
.01
64.2
.60
.75
.32
38.7
.50
2.76
1.03
.66
.64
4.33
.10
53.3
33.5
.54

COMPRESSOR TURBINE COOLING AND LEAKAGE FLOWS FOR NOV/86 TEST

FIGURE 21

UNIT 62 A1

THERMAL EFFICIENCY vs SHAFT POWER



WITH: - I.S.O. AMBIENT CONDITIONS
- 788°C THERMODYNAMIC TURBINE INLET TEMPERATURE DEFINING CAPABILITY

WHERE: POSSIBLE FUTURE MODIFICATION INCLUDES:
- IMPROVED BLADE RING SPRING SEALING
- REDUCED CLEARANCE ON TORQUE TUBE EXIT LABYRINTH SEAL

FIGURE 22

AVON SERIES AND COOPER RT.48S UPGRADE

P.J. HANDY
ENGINEERING SALES MANAGER
ROLLS-ROYCE (CANADA) LTD.
INDUSTRIAL & MARINE DIV.
P.O. BOX 544
MONTREAL, QUEBEC
H4Y 1B3

R.H. MEIER
MANAGER, DESIGN AND DEVELOPMENT
COOPER-BESSEMER ROTATING PRODUCTS
NORTH SANDUSKY STREET
MOUNT VERNON
OHIO 43050
U.S.A.

SUMMARY

THE INDUSTRIAL AVON TYPIFIES AN EARLY DESIGN OF AERO DERIVATIVE GAS GENERATOR WHICH IS RUGGED, RELIABLE AND WELL PROVEN IN THE MARKETPLACE. HOWEVER, WITH CHANGING MARKET FORCES, DEMANDING IMPROVED EFFICIENCY, A NEW CHALLENGE AROSE - NAMELY HOW TO IMPROVE THE AVON'S THERMAL EFFICIENCY WITHOUT AFFECTING ITS MAIN SELLING FEATURES. THIS PAPER SUMMARISES THE DESIGN CHANGES INTENDED TO ACHIEVE SUCH AN IMPROVEMENT APPLICABLE TO BOTH IN-SERVICE AVONS AND NEW PRODUCTION ENGINES.

BACKGROUND

THE INDUSTRIAL AVON WAS ALREADY CLASSED AS A 'MATURE' G.G. IN AERO USE PRIOR TO ITS FIRST INDUSTRIAL APPLICATION IN 1964 AS THE MK.1533 VERSION. INDUSTRIAL EXPERIENCE GAINED IN BOTH LONG TERM GAS FUEL BASE CONTINUOUS OPERATION AND PEAKING DISTILLATE FUELLED OPERATION LED TO COMPONENT IMPROVEMENTS WHICH PERMITTED A POWER INCREASE OF 9%. THIS WAS FOLLOWED BY MATERIAL AND BLADE COOLING IMPROVEMENTS, ALLOWING HIGHER FIRING TEMPERATURES, WHICH CREATED THE MK.1534 AND MK.1535 AVONS IN 1971 AND 1973, AND GAVE 11.5% AND 8.75% POWER INCREASES RESPECTIVELY.

DURING THIS TIME THE EMPHASIS WAS PLACED ON POWER ENHANCEMENT AND IT WASN'T UNTIL THE FUEL CRISIS OF THE EARLY 80'S THAT THE MARKET FORCES BEGAN TO QUESTION THE EFFICIENCY OF THE AVON WHICH, INCIDENTALLY, HAD IMPROVED BY SOME 3.6% LARGELY BY WAY OF INCREASED FIRING TEMPS AT INCREASED RATINGS (SEE FIG.1). THE INTERPRETATION OF THESE MARKET FORCES MEANT THAT A CONVERSION KIT BE MADE AVAILABLE TO UPGRADE EXISTING AVONS. CERTAIN OF THESE FEATURES WOULD ALSO APPLY TO NEW PRODUCTION A1535 ENGINES.

CONSIDERATIONS

GIVEN A CLEAN SHEET OF PAPER, THE POSSIBLE DESIGNS ARE ENDLESS. THE DIFFICULT TASK IS TO SELECT BENEFICIAL FEATURES AND APPLY THEM TO AN EXISTING DESIGN WITHIN THE EXISTING CONSTRAINTS OF PRICE, PROVEN RELIABILITY AND INTERCHANGEABILITY. THE EASIEST CHANGES WITH MINIMAL RISK ARE TO UTILISE INDUSTRIES EXISTING PROVEN FEATURES AND MATERIALS AND ADAPT THEM TO SUIT THE AVON, THEREBY CREATING AN UPGRADE IN TECHNOLOGY WHICH IS EASILY INTRODUCED AT OVERHAUL AND WITH LOW RISK FOR THE OPERATOR. THE FOLLOWING ARE THE DESIGN FEATURES CONTAINED WITHIN THE "EFFICIENCY" CONVERSION KIT AND APPLICABLE TO THE MK.1533 AND MK.1534 AVONS IN SERVICE. THE ENGINE TYPE THEN BECOMES THE MK.1533E AND MK.1534E RESPECTIVELY.

DESIGN FEATURES

ROTOR BLADES -	POLISHING OF ALL TITANIUM BLADES TO IMPROVE EFFICIENCY AND REDUCE DEGRADATION.
STATORS -	POLISHING OF ALL STEEL STATORS.

IN ADDITION, ALL ALUMINIUM BLADES ARE PROTECTED BY SERMETAL W TO PREVENT CORROSION AND CONTAMINATION.

THRUST BEARING - MATERIAL CHANGE FROM VACUUM ARC REMELTED CARBON CHROME (SAE 52100) TO 18% TUNGSTEN, 4% CHROME, 1% VANADIUM (T1), GIVING AT LEAST A TWO-FOLD IMPROVEMENT IN ROLLING CONTACT FATIGUE PERFORMANCE.

THRUST BEARING LOADS - OPTIMISED BY USING THE SEAL ARRANGEMENT COMMON TO THE 1535.

COMBUSTION SYSTEM - OPTIMISED BY USING THE CURRENT 1535 TRIPLE DISH FLAMETUBE.

TURBINE - H.P. TURBINE ROTOR BLADES REPLACED IN INCO 792 - UNCOOLED (FIGS. 2 AND 3).

I.P. NGV REPLACED BY UNCOOLED INCO 738.

P/T - A SELECTION OF P/T MATCHES TO PERMIT EITHER OPTIMUM POWER AND SFC BY UPGRADING AND/OR REMATCHING EXISTING IN-SERVICE UNITS.

PERFORMANCE

THE EFFECT ON EXISTING AVON PERFORMANCE IS INFLUENCED BY

(1) APPLICATION OF THE G.G. MOD PACKAGE.

(2) P/T CAPABILITY TO ACCEPT POWER INCREASE.

(3) P/T CAPABILITY TO REMATCH TO OPTIMISE SFC IMPROVEMENT.

FIGS. 4 & 5 ILLUSTRATE THE BENEFITS TO EFFICIENCY, AS APPLICABLE TO THE AVON 1533 AND 1534 RESPECTIVELY.

HOWEVER, IN ORDER TO ACHIEVE FULL 1535 POWER IT IS NOW PLANNED TO RETAIN THE EXISTING MK.1535 COOLED HP TURBINE ROTOR BLADE. FIG. 6 ILLUSTRATES THE FULL POWER AND EFFICIENCY POTENTIAL AVAILABLE WITH THE OPTIMUM P/T REMATCH.

FIG. 7 SUMMARISES THE OPPORTUNITIES TO IMPROVE SFC AND POWER WITH VARIOUS P/T MATCHES.

THE ABILITY OF A POWER TURBINE TO ACCEPT THE ABOVE POWER/MATCH CHANGES DEPENDS UPON SPECIFIC P/T DESIGNS. AS AN EXAMPLE, ATTACHED TO THIS PAPER IS A BRIEF ADDENDUM ON HOW C-R-I HAVE ADDRESSED TO THESE CHANGES WITH THE RT.48S, P/T.

INDUSTRIAL AVON VARIANTS

THERMAL EFFICIENCY VERSUS EGHP

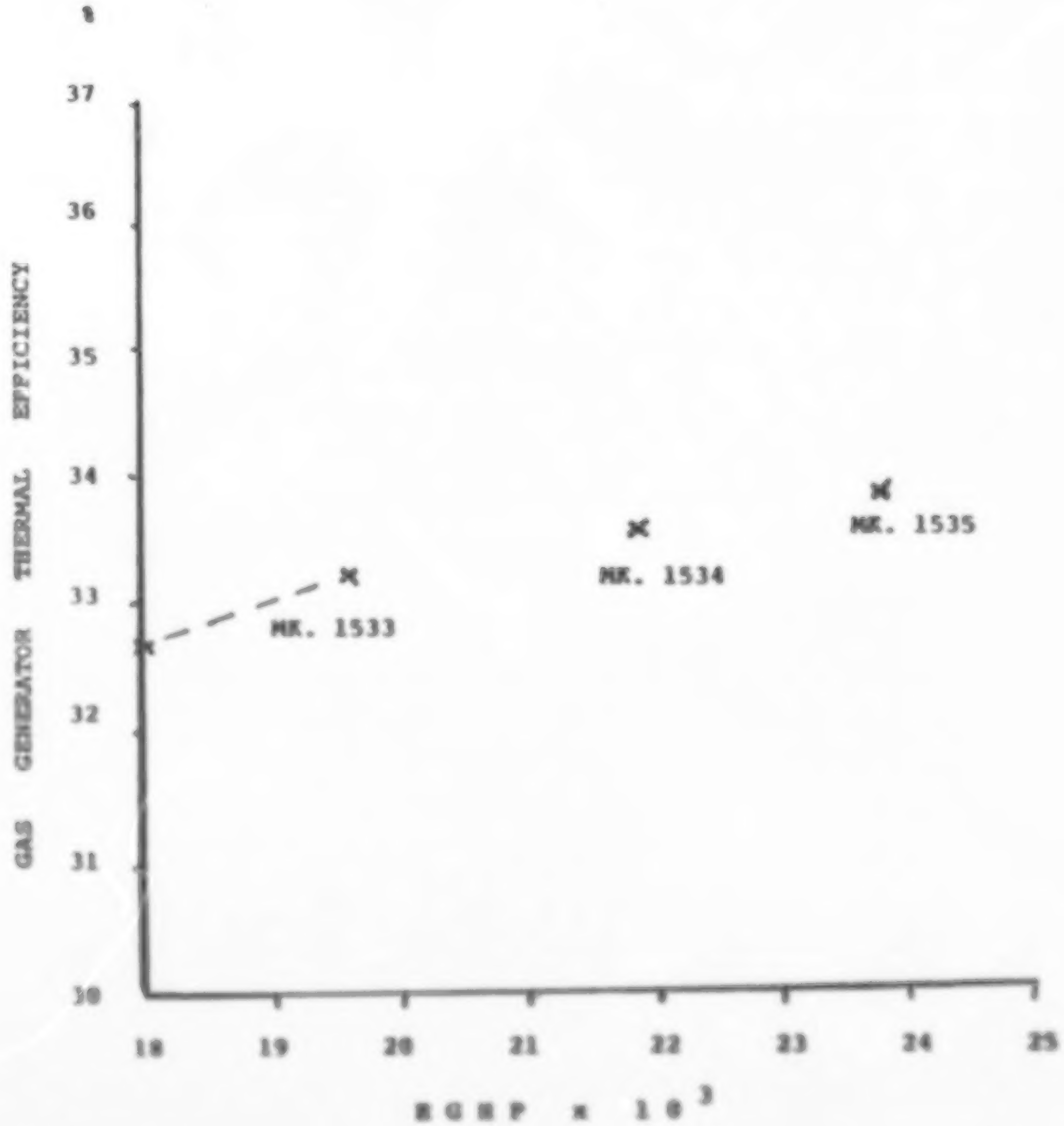
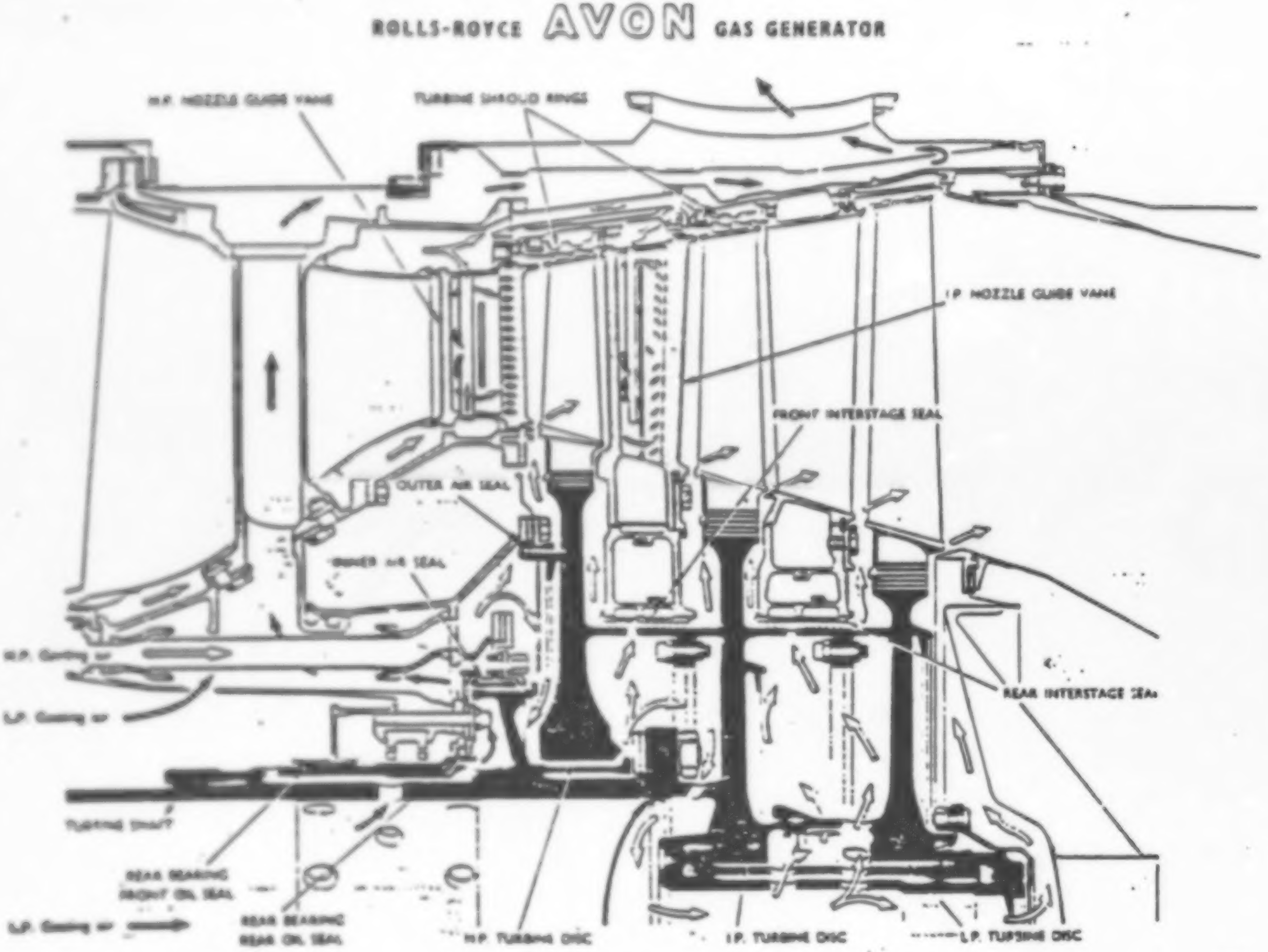


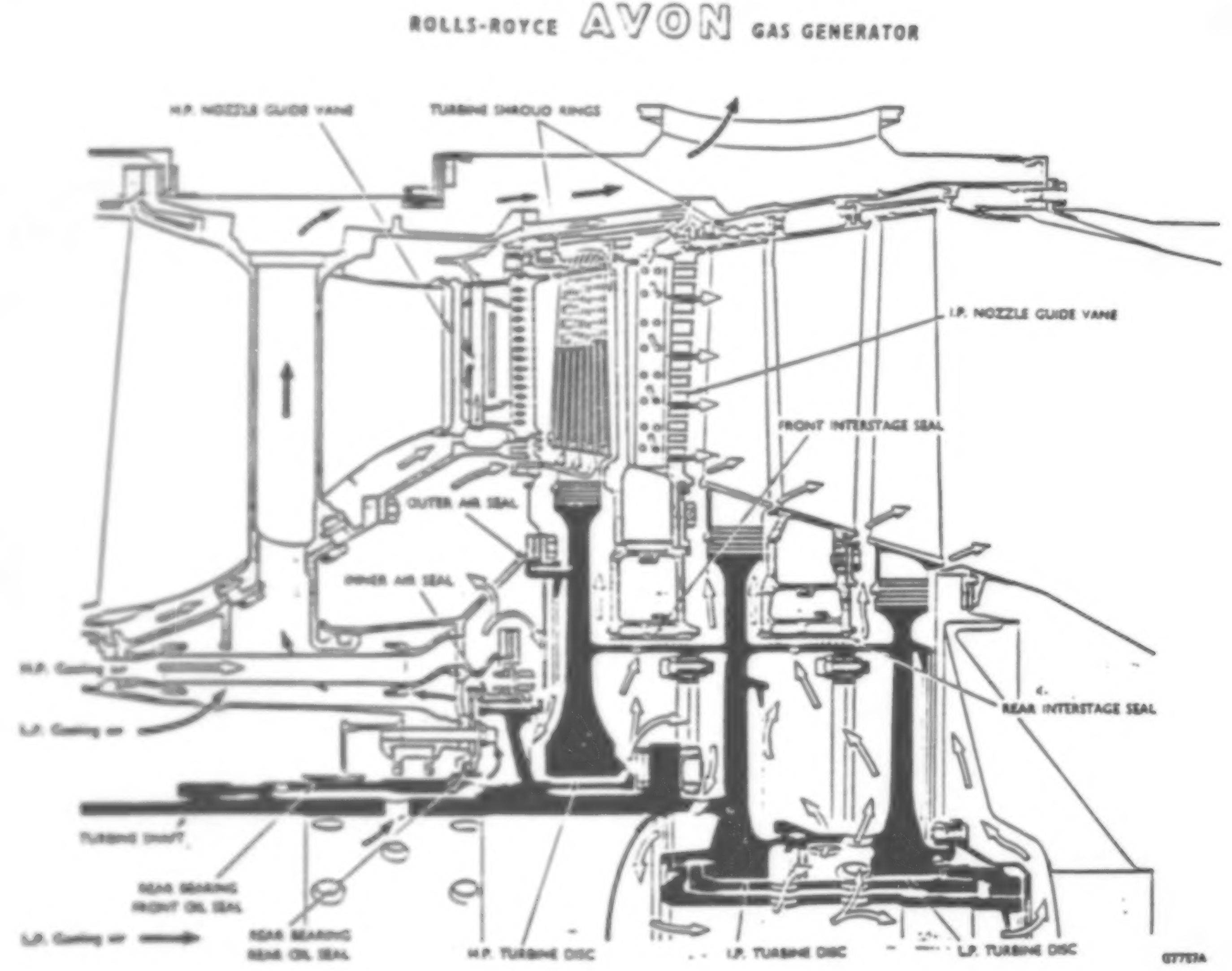
FIG. 1


ROLLS-ROYCE AVON GAS GENERATOR
H.P. NOZZLE GUIDE VANE
TURBINE SHROUD RINGS
I.P. NOZZLE GUIDE VANE
FRONT INTERSTAGE SEAL
OUTER AIR SEAL
INNER AIR SEAL
H.P. Cooling air
L.P. Cooling air
REAR INTERSTAGE SEAL
TURBINE SHAFT
REAR BEARING FRONT OIL SEAL
REAR BEARING REAR OIL SEAL
L.P. Cooling air
H.P. TURBINE DISC
I.P. TURBINE DISC
L.P. TURBINE DISC

1533E AND 1534E SERIES GAS GENERATOR

Turbine cooling air

Fig.2.


ROLLS-ROYCE AVON GAS GENERATOR
H.P. NOZZLE GUIDE VANE
TURBINE SHROUD RINGS
I.P. NOZZLE GUIDE VANE
FRONT INTERSTAGE SEAL
OUTER AIR SEAL
INNER AIR SEAL
H.P. Cooling air
L.P. Cooling air
REAR INTERSTAGE SEAL
TURBINE SHAFT
REAR BEARING FRONT OIL SEAL
L.P. Cooling air
REAR BEARING REAR OIL SEAL
H.P. TURBINE DISC
I.P. TURBINE DISC
L.P. TURBINE DISC

Turbine cooling air - 1534 series gas generator
Fig.3

INDUSTRIAL AVON MK.1533E

DATE

GRAPHIQUE Nº
CHART No.
FIG. 4

RETROFIT INTO MK.1533B INSTALLATION

NATURAL GAS FUEL

MARK NO.	FINAL NOZZLE AREA SQ. INS.	GAS GENERATOR RPM	EXHAUST GAS HORSEPOWER	SFC BTU/EGHP/HR.	EXHAUST CONE TEMPERATURE °C	CHANGE IN SFC %
1533B	344.2	7500	19580	7635	565	0
1533E	344.2	7650	21200	7410	568	-2.9

DATE

GRAPHIQUE Nº
CHART No.
FIG. 5

INDUSTRIAL AVON MK.1534E

RETROFIT INTO MK.1534 INSTALLATION

NATURAL GAS FUEL

MARK NO.	FINAL NOZZLE AREA SQ. INS.	GAS GENERATOR RPM	EXHAUST GAS HORSEPOWER	SFC BTU/EGHP/HR.	EXHAUST CONE TEMPERATURE °C	CHANGE IN SFC %
1534	338.4	7500	21890	7570	630	0
1534E	338.4	7650	22450	7315	601	-3.7

INDUSTRIAL AVON MK.1535 UPGRADE

DATE

GRAPHIQUE Nº
CHART No.
FIG. 6

RETROFIT INTO MK.1535 INSTALLATION

NATURAL GAS FUEL

MARK NO.	FINAL NOZZLE AREA SQ. INS.	GAS GENERATOR RPM	EXHAUST GAS HORSEPOWER	SFC BTU/EGHP/HR.	EXHAUST CONE TEMPERATURE °C	CHANGE IN SFC °C
1535	332.3	7550	23800	7430	630	0
1535 UPGRADE	332.3	7550	23800	7220	625	-2.9

INDUSTRIAL AVON VARIANTS

THERMAL EFFICIENCY VERSUS EGHP

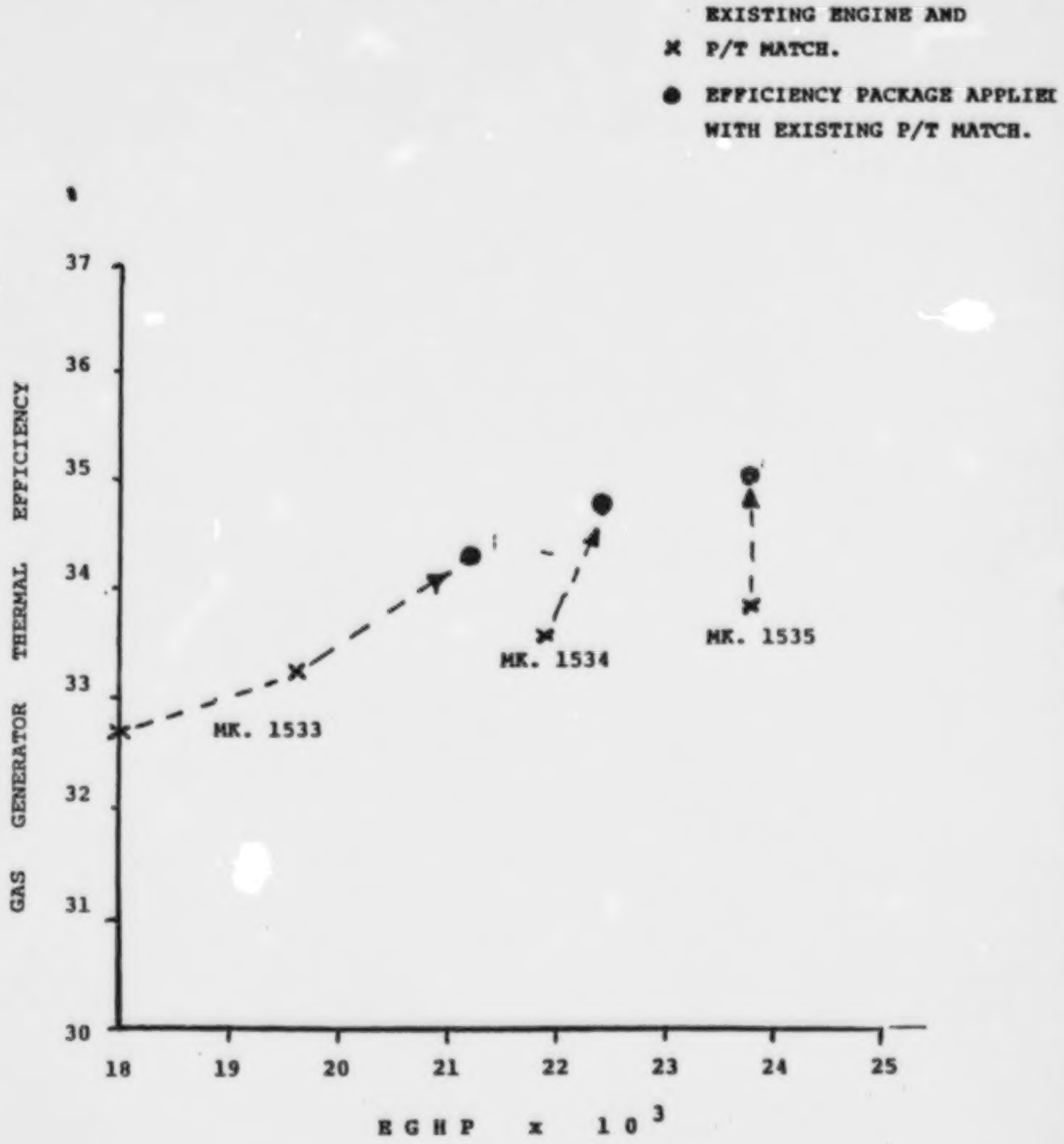


FIG. 7

UPRATING OF COOPER-BESSEMER RT48 POWER TURBINE FOR ROLLS-ROYCE AVON GAS GENERATORS

Robert H. Meier
Manager, Design and Development
Cooper-Bessemer Rotating Products
Energy Services Group
Division of Cooper Industries
North Sandusky Street
Mount Vernon, Ohio 43050

BACKGROUND

The power turbine used with the Rolls-Royce gas generator is model RT48, a two stage free power turbine with 48 inch rotor tip diameter. This design originated in 1959 as the world's first industrial application of an aircraft derivative gas turbine. It was rated at 7820 KW. In 1964 this design was married to the Rolls-Royce Avon in the first North American industrial application of this popular gas generator. A cross-section of the complete gas turbine is shown in Figure 1. The philosophy behind this type of prime mover is a blend of industrial practices with aircraft technology. The hottest part of the cycle and the most highly stressed components are contained in the aircraft derivative portion, which takes advantage of the large body of aircraft development experience and provides an easily serviceable unit. The power turbine by contrast employs the heavier industrial technology and is a permanent part of the package.

DESCRIPTION

Aerodynamically, the power turbine consists of an inlet diffuser to slow the high velocity gases coming from the gas generator, two impulse-reaction stages of airfoils to extract the work, and an exhaust diffuser to reduce back pressure. The rotor blades are mounted by four tooth serrations into individual disks. A lacing wire or damping tube is used in the first stage rotor blade to suppress vibration. The disks are secured to the rotor shaft by through bolts; torque transmission is by precision radial face pins. The power turbine rotor is of an overhung construction, which locates all bearings in a cool area on the drive-end of the unit, right hand side of Figure 2. In this area forced air cooling is not required and the bearings are readily removed for inspection. Stator vanes and casings are supported by struts through the exhaust diffuser. A sliding piston ring type of joint is used to connect the power turbine to the gas generator. All power turbine structures are supported by a heavy cast-iron pedestal on the drive end of the unit.

PERFORMANCE

In response to gains in gas generator performance the output of this unit nearly doubled between 1959 and 1981. This growth is summarized in Figure 3. There have been five major upratings in power turbine design, as indicated in the right hand column of Figure 3. Mechanically these upgrades have involved material improvements to accommodate the higher temperatures, and the addition of disk cooling. Aerodynamically the changes involved primarily first stage stator vane restaggering to match changing gas generator operating lines. Figure 4 gives some more performance specifics related to these upgrades. The gas generator performance was raised thermodynamically by altering its operating line or match to increase cycle pressure ratio and firing temperature. This requires smaller power turbine aerodynamic throat areas. The right-hand side of Figure 4 lists the changes in match area that have been required. The first stage stator nozzle has the most effect on match area and that is where the match adjustments have been made. However, as match area is reduced by restaggering the first stator nozzle alone, efficiency begins to suffer due to unequal work split between stages, reduced stage reaction and increasing airfoil incidence angles. In growing from the original 7820 KW to 15060 KW these effects have adversely affected efficiency by small amounts at first but with increasing amounts in the latest change. By the time of the latest gas generator upgrade the Avon match area had reduced sufficiently that a new look was required into power turbine aerodynamics.

UPGRADE FOR AVON 1535E GAS GENERATOR

Figure 5 summarizes the changes introduced into the RT48 power turbine with this latest upgrade to the Avon gas generator. The new smaller match required a change in the rotor blade airfoils in order to equalize stage work split, restore reaction and raise efficiency. To further raise efficiency the speed was increased slightly, from 5500 RPM to 5775 RPM maximum continuous. Another gain was accomplished by eliminating the damping tube through use of integral interlocking tip shrouds, a design concept borrowed from the larger RT56 power turbine, used with the Rolls-Royce RB211 gas generator. The tip shrouds, in addition to providing a proven method of dampening blade vibration, also provide another efficiency increment by reducing rotor blade tip leakage. As diagramed in Figure 5, the needs for restaggered airfoils and tip shrouds converged into a new blade design. This new blade design is of precision cast form with an integral tip shroud, similar to the RT56 power turbine. The new blades and the higher speed permitted a reprofiling of the disk design into a less bulky shape similar to the disks of the later design RT45 power turbine, used with the Rolls-Royce Spey gas generator. The lighter disks have lower thermal stresses. The more modern RT56 blade-disk three tooth serration form was also incorporated into the rotor. To distinguish this latest model RT48 power turbine with the shrouded rotor, the letter "S" is added as a suffix to the RT48 designation, thus RT48S.

RETAINED FEATURES OF THE UPGRADED RT48S POWER TURBINE

Although the performance improvement in this latest version of the RT48S power turbine is considerable, the basic proven structural design concept is retained. Figure 6 lists the commonalities with the old design. The geometry of the support structures is the same; the rotor-bearing dynamics and bearing accessibility are the same. The stator vanes are the same, the gas generator connection is not altered, and the exhaust system is identical. Figure 7 shows the RT48 power turbine as the modified RT48S. By comparison to Figure 2 it can be seen that the changes are confined to the rotor blades and disks. Existing RT48 power turbines can be converted in situ to the new shrouded RT48S configuration by installation of a new rotor.

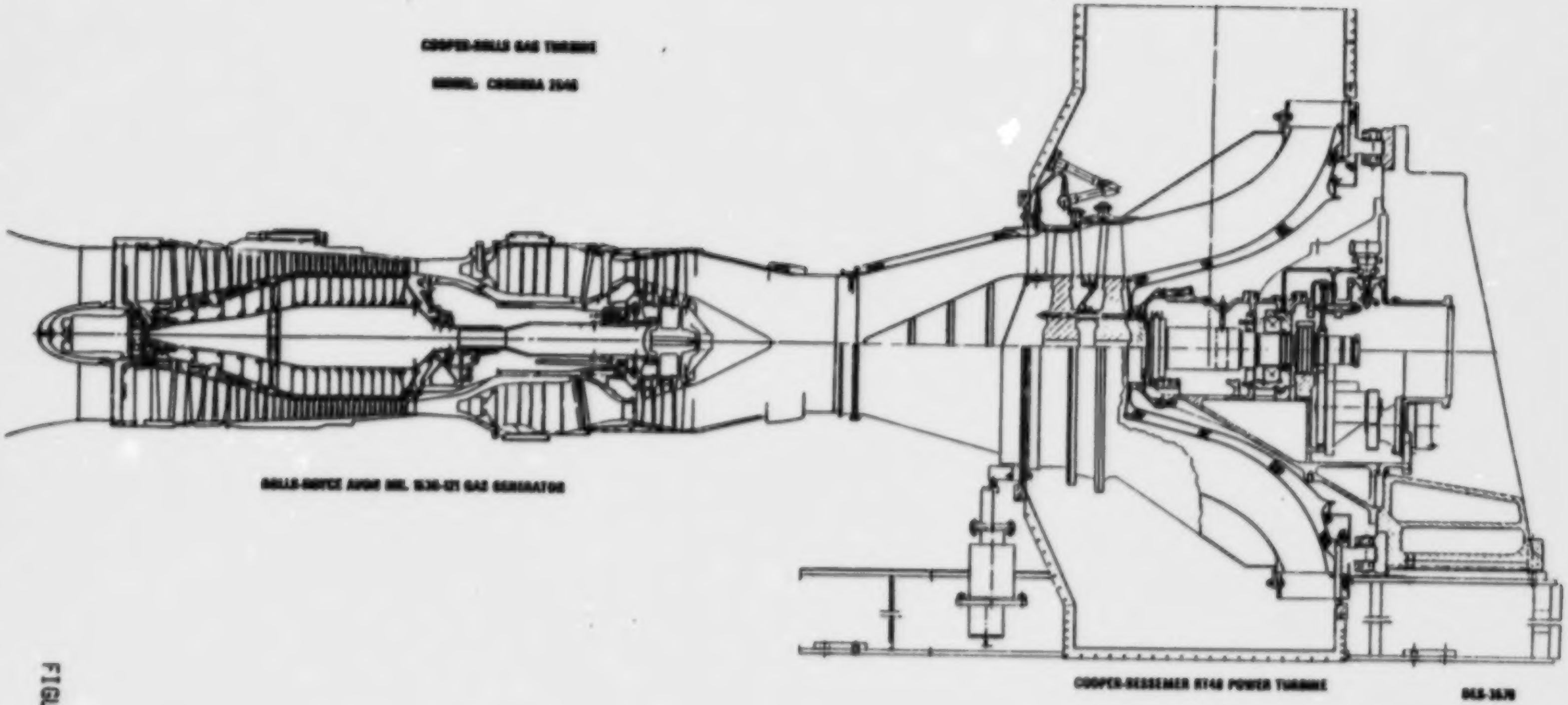
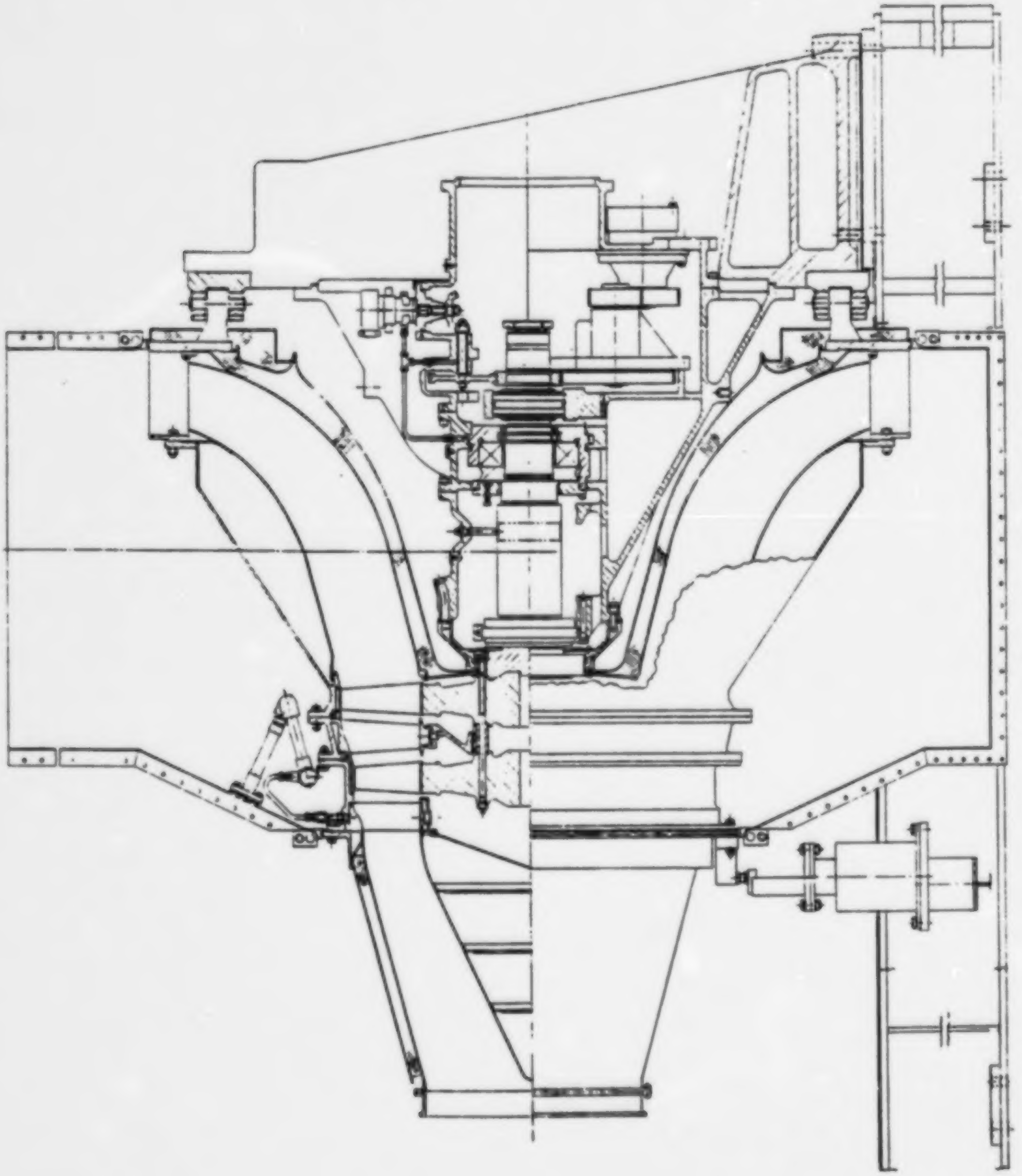

COOPER-ROLLS GAS TURBINE
MODEL: COBERRA 2146
ROLLS-ROYCE AVON MK. 1535-121 GAS GENERATOR
COOPER-BESSEMER RT48 POWER TURBINE
DES-3678

FIGURE 1



COOPER-BESSEMER RT48 POWER TURBINE

FOR ROLLS-ROYCE AVON

DES-3569

FIGURE 2

COOPER-BESSEMER RT48 POWER TURBINE DESIGN HISTORY

YEAR	MODEL	GG	POWER (ISO-KW)	DESIGN CHANGES
1959	RT105	P&WA GG3C-4	7830	—
1964	RT121	AVON 1533-56	10285	RESTAGGER STATOR VANES
1971	2348	AVON 1533-76	12035	RESTAGGER ROTOR BLADES
1972	2448	AVON 1534-101	13420	RESTAGGER STATOR VANES
				ENLARGE EXHAUST
				ADD COOLING
				UPGRADE MATERIALS
1981	2548	AVON 1535-121	14580	RESTAGGER STATOR VANES
				UPGRADE MATERIALS
1987	2648	AVON 1535E-141	15060	NEW SHROUDED ROTOR BLADES
				REPROFILE DISKS
				UPGRADE MATERIALS

JMW185B

15 September 1987

FIGURE 3

COOPER-BESSEMER RT48 UNIT PERFORMANCE HISTORY

GG	POWER (ISO-KW)	MAX. CONT. SPEED (RPM)	THERMAL EFF. (%)	MATCH AREA (SQ. M.)
P&WA GG3C-4	7830	5200	25.4	.243
AVON 1533-56	10285	5200	26.1	.225
AVON 1533-76	12035	5200	27.6	.222
AVON 1534-101	13420	5500	27.6	.218
AVON 1535-121	14580	5500	27.6	.214
AVON 1535E-141	15060	5775	29.7	.210

JMW185C

15 September 1987

FIGURE 4

RT48 UPGRADE FOR AVON 1535E-141

REQUIREMENTS: SMALLER MATCH; IMPROVED EFFICIENCY

MEANS:

- SMALLER MATCH: RESTAGGER ROTOR AIRFOILS
- IMPROVED EFFICIENCY:
  - INCREASE SPEED
  - ELIMINATE DAMPING TUBE
  - ADD TIP SHROUD

NEW FEATURES:

- NEW AIRFOIL CONTOURS
- INTEGRAL INTERLOCKING TIP SHROUDS
- MODERN RT56 SERRATION
- REFINED DISK DESIGN

JMW185D

15 September 1987

FIGURE 5

RT48 UPGRADE FOR AVON 1535E-141

RETAINED FEATURES:	GG INTERFACE
	INLET DIFFUSERS
	STATOR CASINGS
	NOZZLE VANES
	EXHAUST DIFFUSERS
	SUPPORT SCHEME
	BEARINGS
	AUXILIARY DRIVE

JMW185E

15 September 1987

FIGURE 6

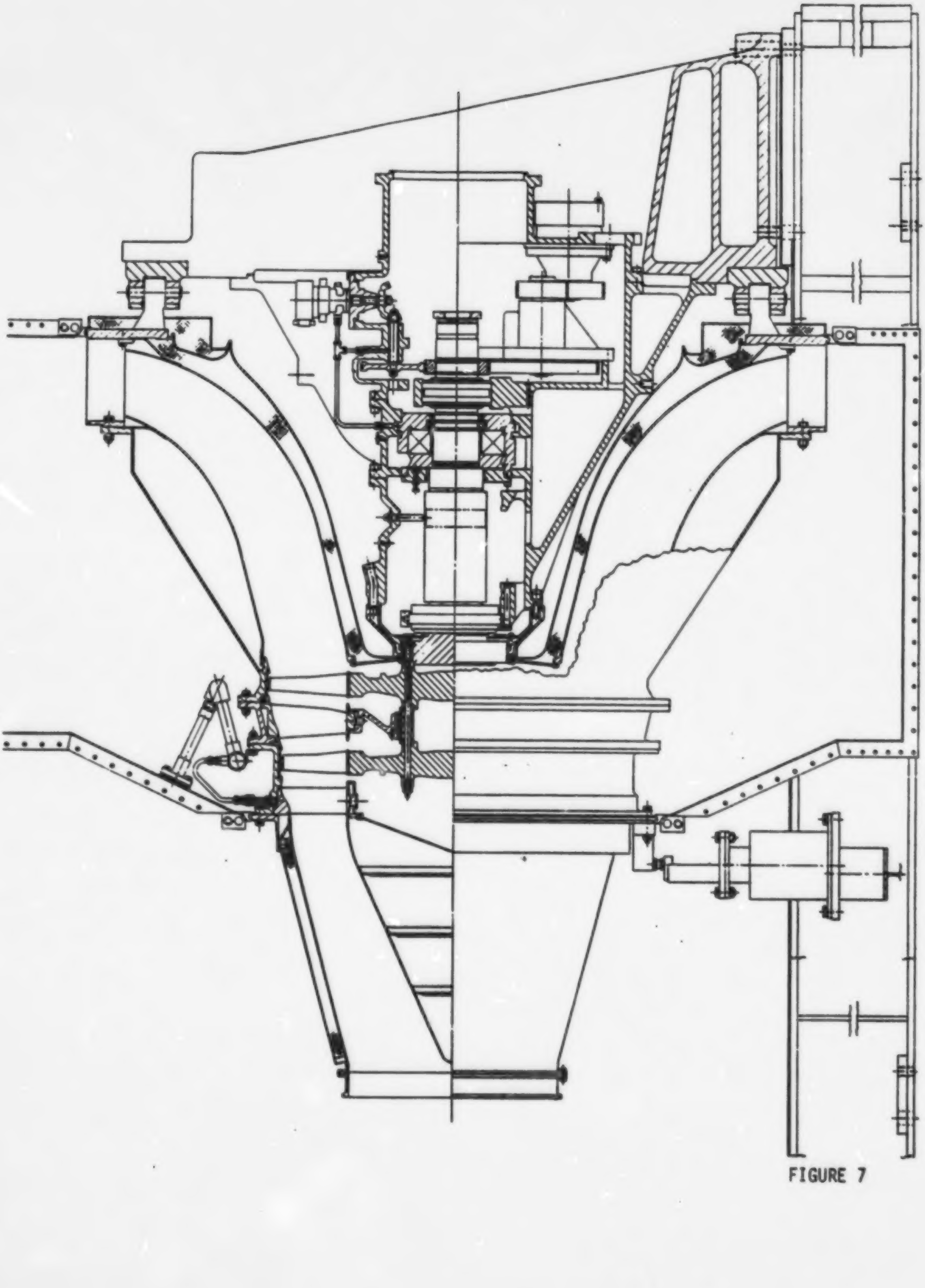


FIGURE 7

BANFF ALBERTA
SEPTEMBER 1987

SEVENTH NRCC SYMPOSIUM
ON
INDUSTRIAL APPLICATION
OF GAS TURBINES

NEW OR REPLACEMENT POWER WITH THE FIRST "THIRD GENERATION" AERO-DERIVED GAS TURBINE

AUTHOR: GORDON N. ROGERS - P. ENG.
DRESSER-RAND

ABSTRACT

The paper discusses the basic philosophy of aero-derived gas turbines and their place in the techno-economic engineering matrix. It introduces to the Symposium the first "third" generation gas turbine, the Dresser-Rand GT-60 which uses the General Electric LM1600 Gas Generator.

INTRODUCTION

Even engineers closely associated with industrial gas turbine applications sometimes lose sight of the basic philosophy of the aero-derivative gas turbine: It is a mechanical, aerodynamically coupled drive train of rotating machinery consisting of a core aircraft engine, producing hot gas which is expanded through an industrial power turbine. The key always has been, and always will be, that aero-derived industrial gas turbines and their users, piggyback the aircraft engine industry, which receives extensive Government funding to achieve high efficiency and high specific power for military aircraft engines. The power turbine machinery manufacturers, and their customers, benefit by essentially only being required to fund the development of the remainder of the drive train. Whereas thousands of aircraft engines are produced, only hundreds of gas turbines are required for mechanical drives, and yet these too, benefit from the funding base of many millions of dollars.

Experience is gained in the aircraft industry - capitalizing on development dollars, and the cost of flying hours to provide an experience base which is commercially impossible in the industrial market.

The GT-60 is the first "third generation" aero derived gas turbine.

THE F404 FIGHTER TURBOFAN ENGINE

The F404 fighter turbofan engine was competitively selected by the U.S. Navy for the F/A-18 fighter/attack aircraft; it first entered service in 1978, eight years after its development began.

One of the key design criteria established by the U.S. Navy for this competitive selection was reliability. The F404 has demonstrated success against this criteria by achieving a 200% - 600% increase over current fighter engines, much of this success being attributed to a significant decrease in the total number of parts.

THE LM1600 GAS GENERATOR

The LM1600 gas generator is derived directly from the F404 with only minimal modifications (see Fig. 1). The three "fan" stages of the low pressure compressor are "clipped", thus reducing the flow path diameter approximately four inches; also, the bypass duct which is redundant in the gas generator versions, is eliminated.

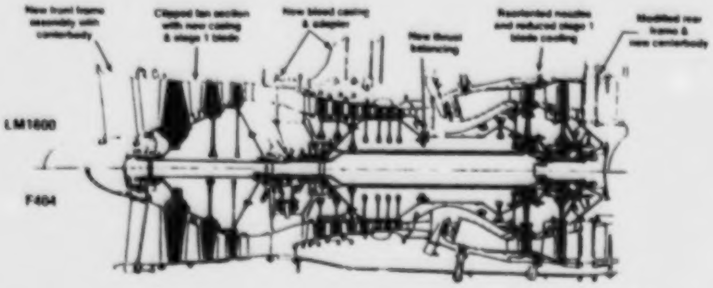

New front frame assembly with centerbody
Clipped fan section with new casing & stage 1 blade
New bleed casing & adapter
New thrust balancing
Reoriented nozzles and reduced stage 1 blade cooling
Modified rear frame & new centerbody
LM1600
F404

FIGURE 1

A new front frame assembly was introduced which industrializes the aircraft engine version. A bellmouth and a stationary forward center body were also added for straightening and guiding the inlet air into the engine. The aluminum bellmouth contains a water wash manifold and forms the outer flow path contour into the front frame; the aluminum forward center body attaches to the hub of the front frame and forms the inner flow path contour into the front frame.

The LM1600 derivative retains all the F404 latest technologies of advanced cooling, corrosion-resistant materials and coatings, high loading, low aspect ratio blade design, and modular construction.

The LM1600 gas generator contains only ten stages of compression:

> The 4:1 pressure ratio 3 stage (with variable inlet guide vanes) axial flow low pressure compressor (LPC), is driven by a single stage low pressure turbine.
>
> The 5.5:1 pressure ratio, 7 stage high pressure axial flow compressor is driven by a single stage high pressure turbine. The high pressure compressor is identical to that in the F404.

The stator case assembly consists of a titanium forward section which houses the variable inlet guide vanes, variable stages 1 and 2, and fixed stage 3 vanes, and a high temperature nickel-base alloy second section with the fixed vane stages 4 through 6. The combustor is an annular flow-through type of the same design as the F404. The low and high pressure turbines are identical to the F404 except for minor changes for industrialization such as nozzle flow area reductions for better component matching; both are fully air cooled.

The simplicity of the LM1600 is best shown by Fig. 2 - an illustration of the two-rotor configuration and bearing arrangement.

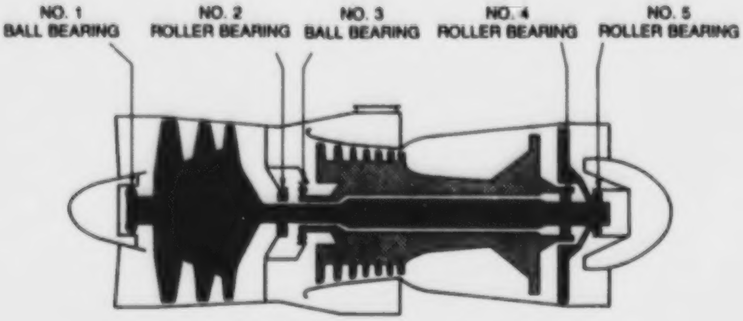


FIGURE 2

The base introductory rating of the LM1600 firing temperature will be 2210 deg. F (1210 deg. C) - considerably lower than the F404 take-off value.

THE GT-60 GAS TURBINE

The power turbine is derived directly from the Dresser-Rand E-232 fluidic catalytic cracker unit (FCCU) power recovery expander. The same blade tooling is used but to optimize efficiency the second stage stator has been re-designed. Some material changes were necessary for compatibility with the gas generator, and some features unique to the FCCU application were changed (see Figure 3).

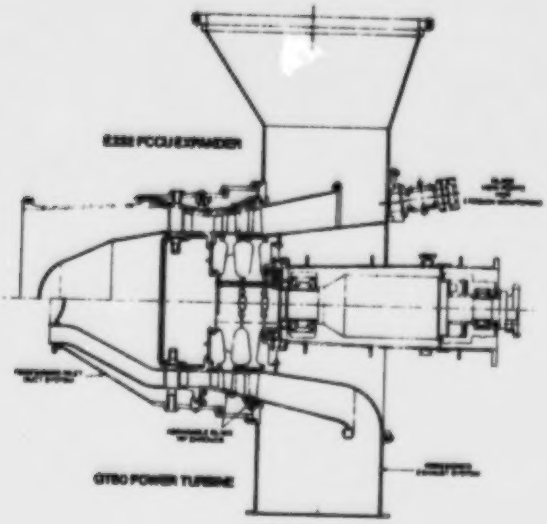


FIGURE 3

This is an excellent match for the LM1600 and provides a continuing genealogy. In 1970 the GE LM2500, derived from the TF39 and CF-6 family of "second generation" engines was introduced in the 24,000 - 28,000 BHP class with a greatly improved thermal efficiency from that of first generation gas turbines. The gas generator was coupled with one of Dresser-Rand's power turbines to make the GT-61 gas turbine. The technology of the time drew on experience gained in FCCU power recovery units. For example, the Ingersoll-Rand E-148 FCCU power recovery train installed in an oil refinery in 1966 was rated at 15480 BHP with a gas stream of 1200 deg. F. In comparison, one of the most commonly used first-generation gas generators at that time, the Rolls Royce Avon had an exhaust gas temperature of only 1024 deg. F. This high temperature technology was, and is, directly applicable to the higher temperatures of second and third generation gas turbines.

The power turbine for the GT-60 gas turbine is a two-stage, multiple tiebolt, cantilever rotor configuration with an axial inlet and radial exhaust typical of aero-derivative gas turbine power turbines. It is bolted rigidly to the gas generator exhaust flange forming a single, unitized, prime mover. The main support trunnions are located on the power turbine stator casing in line with the first stage stators. The gas generator and the power turbine grow axially from this point. There is a forward mount on the gas generator at the compressor mid frame. The rear power turbine support is a cantilevered horizontally split pedestal that houses the rotor and bearing. There are no struts in the exhaust diffuser. This configuration minimizes the losses in the exhaust system, particularly at part loads. The stator and rotor thrust loads are transmitted through the structural supports to the baseplate.

As shown in Figure 3, the inlet casing assembly incorporates a double-wall construction to reduce thermal gradients in the outer casings. Insulation is used both externally and internally to optimize temperature distribution, minimize distortion, and increase the low cycle fatigue life of the hot structural parts. The outer casing bolts directly to the interface flange of the gas generator. The inner liner forms the inner wall of the flow path, and guides the hot gas from the gas generator to the first stage of the power turbine; both the inner and outer walls of this liner mate with finger seals attached to the exhaust liner of the gas generator.

A pressure head is bolted to the upstream end of the inner wall to form a tight seal and prevent bypass leakage of the hot gas. The outer wall of the liner is clamped to the outer casing at the downstream end and supported by radial pins that permit unrestricted radial differential expansion. The inner wall of the liner is supported to the outer wall by seven hollow, airfoil struts just upstream of the first stage stator. The exhaust diffuser and casing have been aerodynamically designed, including the addition of contoured inner fairings inside the exhaust case, to control velocity distribution, and achieve maximum pressure recovery between the blading and exhaust flange.

The rotor is supported by two tilting pad journal bearings, each of which have five pads with load between pads. Active thrust is contained by a 12 inch tilting pad thrust bearing, and a smaller inactive tilting pad thrust bearing, both with equalizing rings. The shaft has a rather large center body to provide the desired bearing load distribution and enhance the rotor dynamics. The discs are attached to the shaft by six tiebolts.

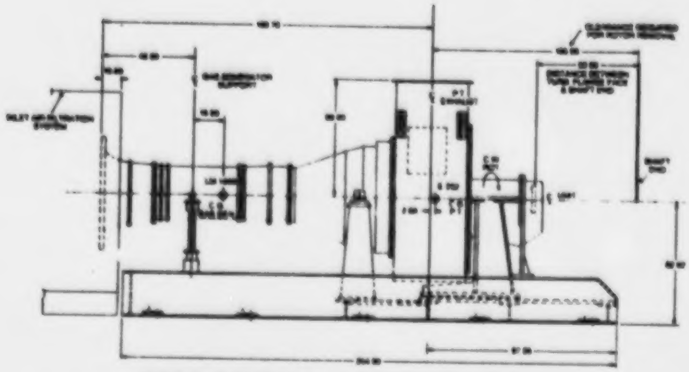
Radial alignment is established and maintained by rabbet fits between discs and between the second stage disc and the shaft. Radial pins at the splitlines between each rotor element provide torque-carrying backup for the friction force exerted by the tiebolt preload.

Rotor blades are assembled in axial fir tree slots, broached at an angle, and locked with a wire to prevent axial movement.

The first stage, investment cast, stator vanes are cantilevered from an inner, unsplit shroud ring; a spindle on one end of the vane fits in a hole in the ring and is locked at the proper angle by a key that fits in a slot on the spindle.

The second stage stator is fabricated with six investment cast vanes in each of ten segments. A flange on the segment outer shroud is bolted to the unsplit, stationary shroud ring of the second stage rotor blade. Nicroseal-filled honeycomb is integral with each stationary shroud in line with the rotor blade tips.

THE GT-60 GAS TURBINE PACKAGE



GT60 GENERAL ARRANGEMENT

FIGURE 4

Together, the LM1600 gas generator and the power recovery turbine form the Dresser-Rand GT-60; it is a simple cycle two shaft gas turbine designed in accordance with API standards for industrial applications. A typical arrangement is shown in Figure 4.

The gas turbine driver skid can be furnished with either a mini-baseplate arrangement or a full baseplate; with either one, the gas generator auxiliaries are packaged in one of two modules which are just off the skid or in enclosure recesses on each side of the engine, accessible from outside the enclosure.

One contains the synthetic oil reservoir, the filters, main valves, controls, and instrumentation associated with the synthetic lube system. Interconnecting piping and wiring runs from there to the gas generator, oil cooler, switch and gauge panel, and control panel, as required.

The other module contains the systems associated with the fuel management and emission control systems, with the exception of the fuel control valve which is mounted close to the engine. Otherwise, only interconnecting wiring and piping runs to the engine or panels. This arrangement keeps the skid area near the gas generator clutter-free, and for those units with enclosures, these systems are accessible without entering the enclosure. Another advantage is these modules can be standardized and used with minimum changes for either the mini-base or full baseplate arrangements.

The inlet and exhaust systems are furnished to meet the application requirements, and the exhaust collector can be rotated 60 degrees either side of vertical.

The gas generator is bolted directly to the power turbine transition casing and supported by forward mounts on the compressor mid frame. The power turbine is supported on a small base which can be integral with a larger, full-size, baseplate, or mounted separately on customer's soleplates, in a mini-baseplate configuration. The power turbine requires a drive coupling spacer length of about 50 inches to permit moving the rotor and bearing support pedestal far enough forward of the driven equipment to remove the rotor, without requiring removal of the gas generator or exhaust system. The driver skid extends beyond the end of the power turbine by about the same amount. The coupling can either be diaphragm or continuously lubricated gear type.

The driver skid is 22 feet long by 8 feet wide. The weight of the driver skid with a full baseplate, including auxiliaries, is about 35,000 lbs. The GT-60 footprint is invariably smaller than that of older equivalent units.

The performance of the GT-60 at ISO conditions is as follows:

Power	16,600 HP (12,380 KW)
Heat Rate	7,200 BTU/HP-HR (10188 KJ/KW-HR)
Exhaust Mass Flow	96 LB/SEC (43.5 KG/SEC)
Exhaust Temperature	882 deg. F (472 deg. C)
GG Pressure Ratio	21.5:1
GG Firing Temperature	2210 deg. F (1210 deg. C)
Power Turbine Speed	6800 RPM

APPLICATIONS

Obviously this first "third generation" gas turbine is the best in its class for new installations but its specific fuel consumption is so good compared to first generation units that there are many opportunities to replace existing installations with economic benefits. Many factors contribute to a complete evaluation and every pipeline - indeed every compressor station - is different. However, taking the simplest approach: at $3.00 per million BTU of fuel gas a 16,000 HP gas turbine operating at full load ISO conditions for 8,000 hours per year, will consume fuel approximately as follows:

First generation (average 9790 BTU/HP-HR)	$3,759,360 per year
Third generation (average 7200 BTU/HP-HR)	$2,764,800 per year
Fuel savings are therefore about	$1,000,000 per year

However, this is not the complete story; using fuel savings to initiate a specific study will expose many possibilities for additional economic benefits.

Some pipelines are upwards of thirty years old. Frequently as their capacity grew they were powered in relatively small increments in the early years. The resulting installations may have an incredibly high overall fuel usage per MMSCFD of gas for a given compressor station pressure ratio when measured from mainline connection to mainline connection, for reasons including:

Existing compression equipment may not be operating at or near the original design point.

Smaller gas turbines and compressors are inherently less efficient.

Old equipment may be nearing the end of its useful life unless major parts are replaced.

Replacement spare parts, even if available, may be very expensive.

Multiple unit stations have high station piping pressure losses.

Existing inlet and exhaust systems may produce high losses.

To these economic factors consideration must be given to the savings available from all the technological developments which have occurred since the original equipment was installed. Examples include:

State of the art inlet systems keep the gas turbine compressor clean and free from fouling resulting in higher efficiency and lower maintenance.

State of the art control systems suit present day company infrastructure, supervisory requirements, and provide improved monitoring for lower operating costs.

Current pipeline compressor design uses CADAM systems to design and manufacture compressor impellers. Historically, impellers were designed manually and manufactured by forming and welding blades. These procedures limited the designers' degrees of freedom as well as the accuracy of manufacture. New designs are generated to very close tolerances of contour manufactured on computer operated machine tools. The resulting impeller conforms to the design of the most complex blade shape.

Similar improvements are available for diffuser components.

Dry gas seals not only reduce parasitic losses and downtime outages but avoid seal oil contamination of the pipeline for improved efficiency.

Magnetic bearings further reduce parasitic losses and operational costs. Several papers have been published on the subject and the consensus is positive for new machinery.

New electronic surge controls can result in significant fuel savings and better surge protection.

Taking advantage of all these benefits it is possible to reduce specific fuel consumption (fuel used to compress gas) at some older locations by as much as fifty percent with correspondingly large improvements in outages, downtime and operational costs.

FUTURE OF THE LM1600

The F404 engine is now used worldwide for the U.S. Navy, the Canadian and Australian F/A-18 programs and selected for Spanish, Swedish and French applications. In 1987 the F404 engine family has exceeded one million hours total flying time, an experience base which helps to assure the high reliability achievable using the LM1600 core gas generator.

Current production schedules for the F404/LM1600 extend beyond the year 2000 - a significant fact when selecting replacement machinery.

CONCLUSION

Introduced at 16000 - 18000 HP and with the highest efficiency in its class the GT-60 using the GE LM1600 provides a great opportunity to economically repower or replace many existing older gas turbine installations.

CONVERSION OF GAS COMPRESSORS TO IMPROVE EFFICIENCY

Robert H. Meier
Manager, Design and Development
Cooper-Bessemer Rotating Products
Energy Services Group
Division of Cooper Industries
North Sandusky Street
Mount Vernon, Ohio 43050

ABSTRACT

The application of a well proven design concept to achieve superior aerodynamic efficiency in new centrifugal pipeline boosters to existing units is presented. Its evolution, the extent of effort required for implementation and effectiveness are discussed to help evaluate the benefits of such a conversion in any specific operating environment.

INTRODUCTION

Development of turbomachinery for industrial service has consistently focused on efficiency improvement. Whether this led to new concepts, larger units, more modest upratings of existing units or only improvements to individual components of an existing machine, each development step had efficiency improvement as its single main objective. In this development the Canadian mainline gas transmission companies have played a pioneering role: Their foresight and unique operating environment permitted them to introduce prototype concepts into commercial service to help these units accumulate the operating experience so essential to their general acceptability to even the most conservative users. This active interest in progress has in some instances led to the direct promotion of new approaches with other users or the pursuit of new ideas with equipment suppliers to realize an objective despite a sometime lack of enthusiasm on the supplier's side. Some examples of this pioneering spirit follow the development history of turbomachinery over the last thirty years: Introduction of first and second generation aero derivative gas turbines, periodic aerodynamic restaging of centrifugal pipeline boosters, axial inlet boosters, dry gas seals, magnetic bearings, etc.

It is through this active involvement in turbomachinery development over more than two decades and due to the strong cost pressures exerted by the existing market conditions that TransCanada Pipelines found a unique solution to an increasing efficiency problem in many of their oldest centrifugal pipeline boosters. Since their original installation in 1964 these units experienced a gradual increase in throughput; the resulting increase in transmission losses was overcome with a combination of line looping and addition of new compression stations. These changing conditions were accommodated by periodic replacements of the aerodynamic assemblies in the boosters, which returned them to operating at optimum efficiency levels. Typically, the throughput over twenty years more than doubled, from originally 850 to over 1850 MMCFD while the pressure ratio dropped from nearly 1.45 to 1.18 or translated into head from 47.2 to 22.5 KJ/kg. This combination of flow increase and reduced head requirement led to the dramatic changes in aerodynamic configuration shown in Figures 1 and 2: The original two stage assembly of moderate flow capability (Figure 1) progressively evolved into the much higher flow single stage design shown in Figure 2.

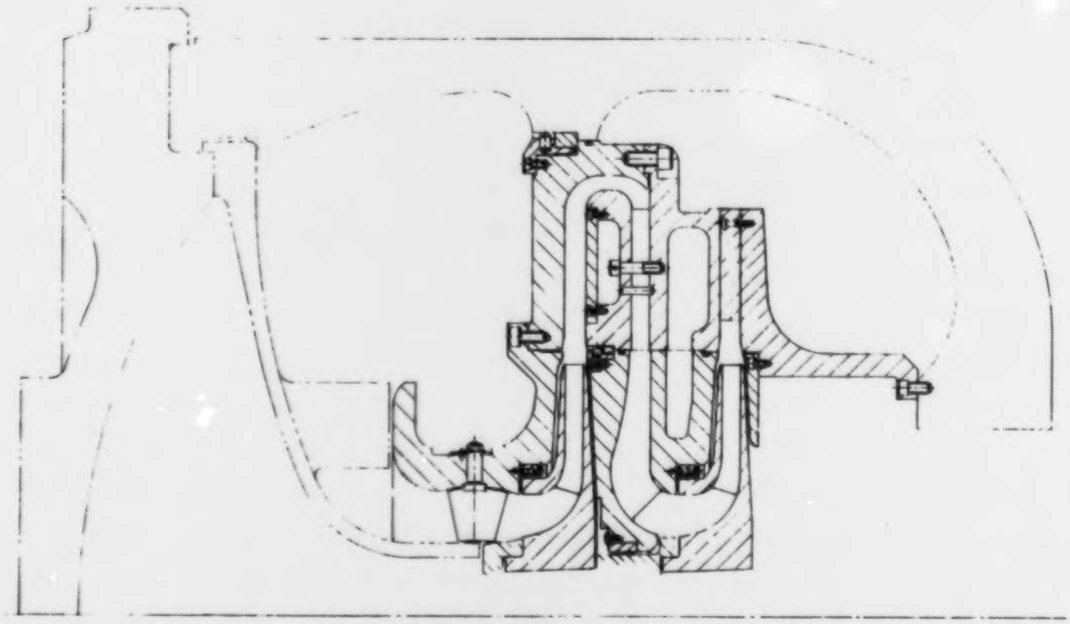


FIGURE 1 ORIGINAL AERODYNAMIC CONFIGURATION

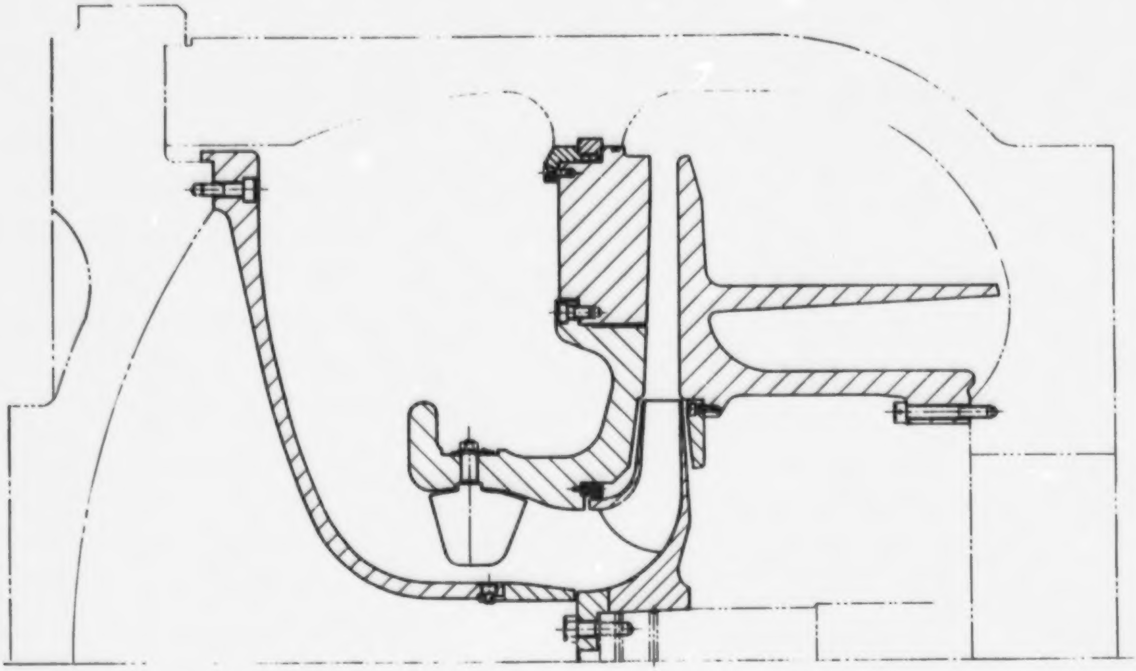


FIGURE 2 PRESENT AERODYNAMIC CONFIGURATION BEFORE CONVERSION TO AXIAL INLET

The large increase in flow exceeds good design practice for flow velocity in piping and the compressor itself. This translates into significant losses (pressure drops), which directly affect fuel consumption. This recognition, and the positive experience with highly efficient axial inlet boosters, encouraged TCPL to pursue the feasibility of converting existing boosters with a conventional side inlet to an axial inlet. At the same time they increased the unit piping to a larger size to reduce piping losses.

This paper will be limited to the compressor conversion itself and only address the modifications to the surrounding equipment to the extent directly necessitated by the compressor modifications.

CONVERSION CONCEPT

From the beginning it was recognized that the total conversion costs had to be kept to absolute minimum to assure a full payback through fuel savings in no more than two years. This objective limited the potential performance gains significantly as it prevented the outright replacement of the existing compressor with a true axial inlet booster. The effects of this compromise on performance and maintainability will be reviewed later on.

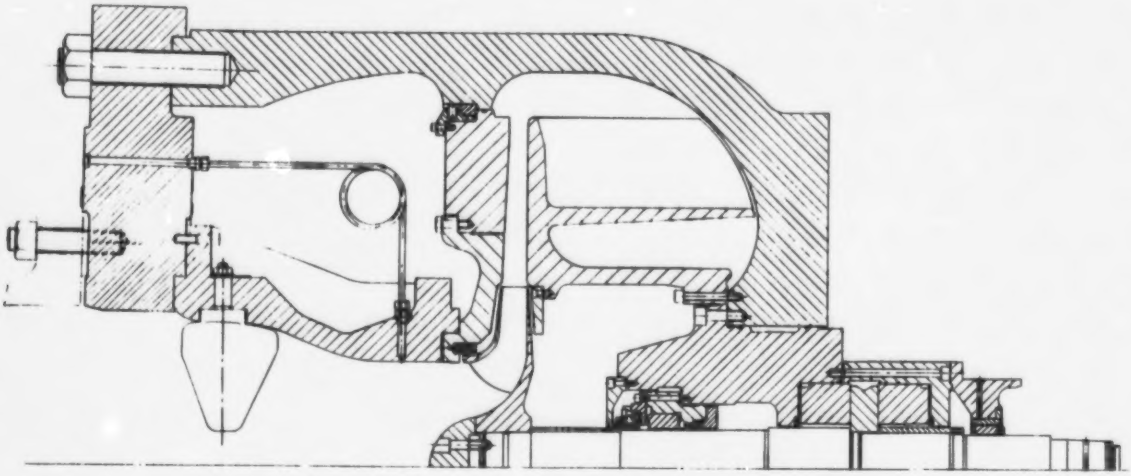


FIGURE 3 COMPRESSOR CROSS SECTION AFTER CONVERSION TO AXIAL INLET

The overall conversion effort requires modifications to the compressor itself and its suction piping. A comparison of Figures 2 and 3 illustrates the conversion impact on the compressor: The main new compressor component is a casing cover, which accommodates the new axial intake flange including its retaining studs; it also supports a new intake transition piece, which ducts the flow from the intake flange into the impeller eye, carries the new inlet guide vanes and seals-off the original intake scroll. At its downstream end this transition piece interfaces with a remachined inner intake wall that also positions the original eye seal labyrinth. An aerodynamically shaped impeller retaining nut completes the list of new components required besides the hardware necessary for retaining the new components and sealing to contain the gas. Note that all other components and assemblies within the compressor are left unchanged: This includes the impeller and discharge volute plus the complete rotor, bearing and seal assemblies.

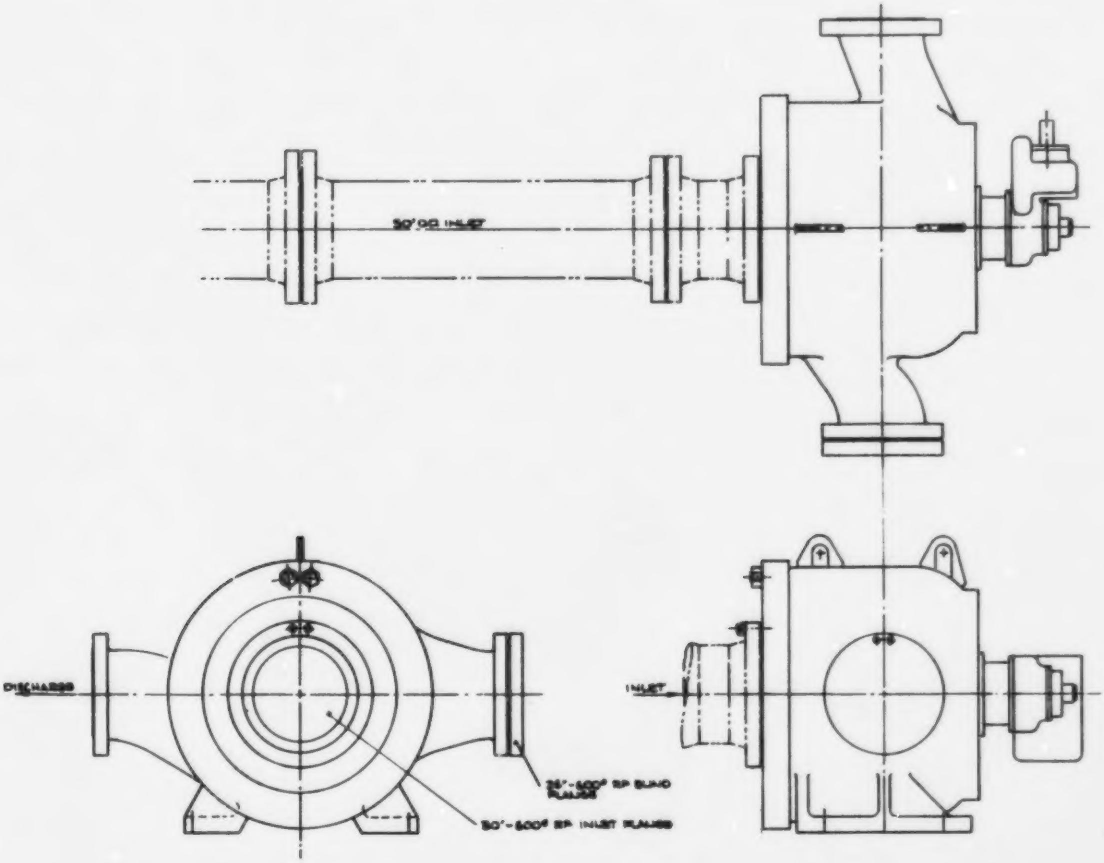


FIGURE 4 COMPRESSOR PIPING ARRANGEMENT AFTER CONVERSION

Externally, the original suction flange is simply blanked-off with a blind flange as shown in Figure 4. To satisfy safety concerns, the converted compressor should be exposed to a gas leak test prior to recommissioning, and the new cover hydrostatically tested at the end of its manufacturing cycle.

The main conversion effort involves the relocation of the suction piping also shown in Figure 4. To maintain the service access to the aerodynamic, rotor and seal assemblies in the converted booster, the new suction pipe contains two spool pieces: The downstream section is used to accommodate the suction flange studs and counterbalance the new cover carrying the intake transition piece; it should never require separation from the cover. The upstream spool piece provides the direct maintenance access to all compressor internal components.

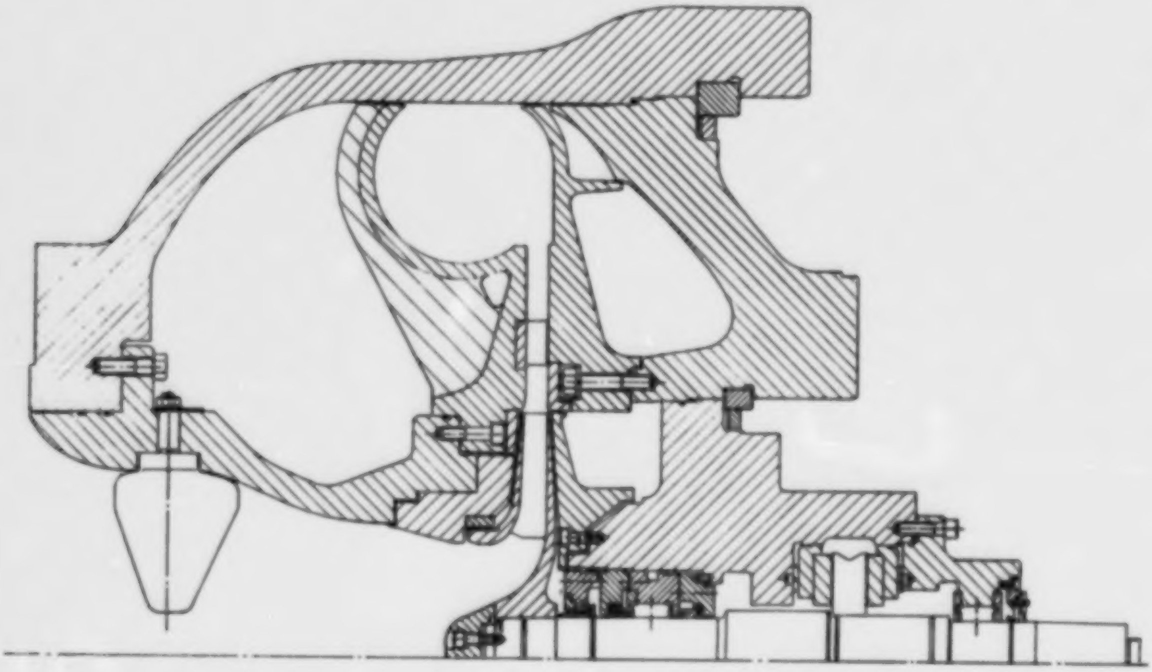


FIGURE 5 TRUE AXIAL INLET PIPELINE BOOSTER - RFA36

Maintainability is one of the two major areas in which an existing pipeline booster converted to axial inlet differs from a true axial inlet booster such as the RFA36 shown in Figure 5. The latter is designed for complete service access from its drive end by arranging its end cover facing the driver. This permits complete maintenance including an aerodynamic changeout without disturbing the suction pipe; this service access is further simplified through the provision of an inner bundle, containing all moving elements plus their supporting and sealing components, which is removed without disturbing the much larger end cover. On a converted conventional booster, however, the suction pipe spool pieces and end cover will need to be removed for most maintenance work.

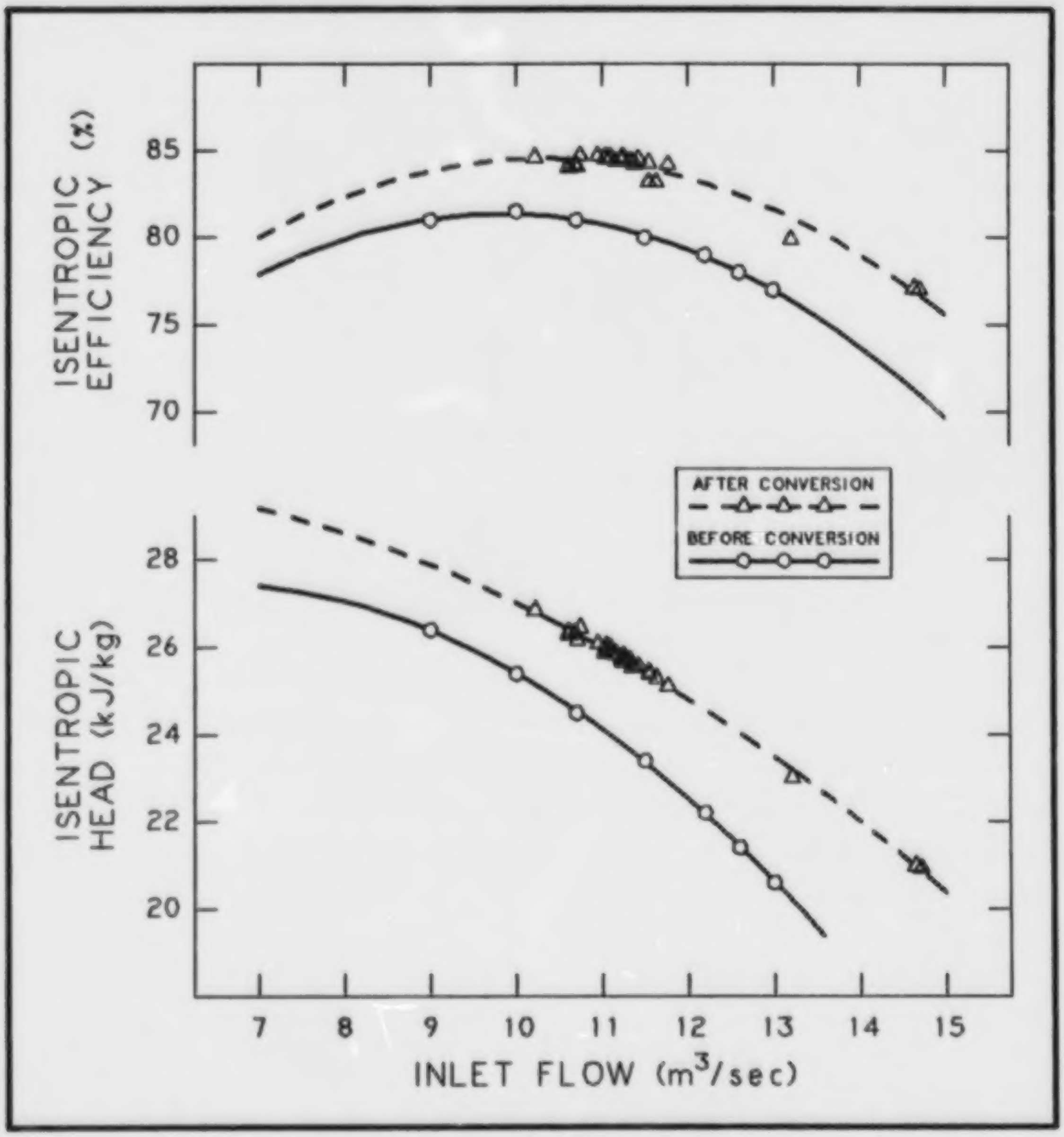


FIGURE 6 ACTUAL AERODYNAMIC PERFORMANCE TEST RESULTS BEFORE AND AFTER CONVERSION

EFFECT ON PERFORMANCE

Since the sole justification for this compressor conversion was its positive effect on aerodynamic efficiency and its resulting fuel savings, the careful measurement of the compressor performance before and after the conversion was essential to judge the success of the program. This testing was carried-out by TransCanada Pipelines. Excellent consistency of results over the full operating characteristic on every unit converted was achieved. Figure 6 shows a comparison of test results taken before and after the conversion to axial inlet. While all the test results shown here were measured on the same unit, they are typical of all units converted to date. The graph confirms an isentropic efficiency improvement at optimum flow from 81.5% before to 84.7% after the conversion; a gain from 73.9% before to 79.65% after was achieved at 130% of optimum flow. This performance improvement is based on flange to flange measurements (static) and consequently excludes the gains from the reduction of transmission losses realized from simultaneously increasing the size of the suction pipe.

It should also be noted that the installation of an RFA36 would have permitted a further efficiency improvement to near 87.0%. This additional gain would have been realized through the larger booster frame size resulting in generally lower gas velocities, a more efficient impeller design (improved flow and tolerance control), a carefully tuned vaneless diffuser and a closely matched roll volute. Also, the installation of a new true axial inlet booster of larger frame size would have assured the retention of the highest efficiency levels well into the future through several additional flow increases.

Figure 7 was developed to help evaluate the potential effects of an axial inlet conversion on any existing centrifugal pipeline booster. It shows (clearly) that the effectiveness of such a conversion depends on the actual volume flow rate to be handled relative to the booster frame size used, and on the head requirement of the aerodynamic assembly involved. The higher the relative flow rate, and the lower the head requirement, the greater gain in efficiency improvement is expected.

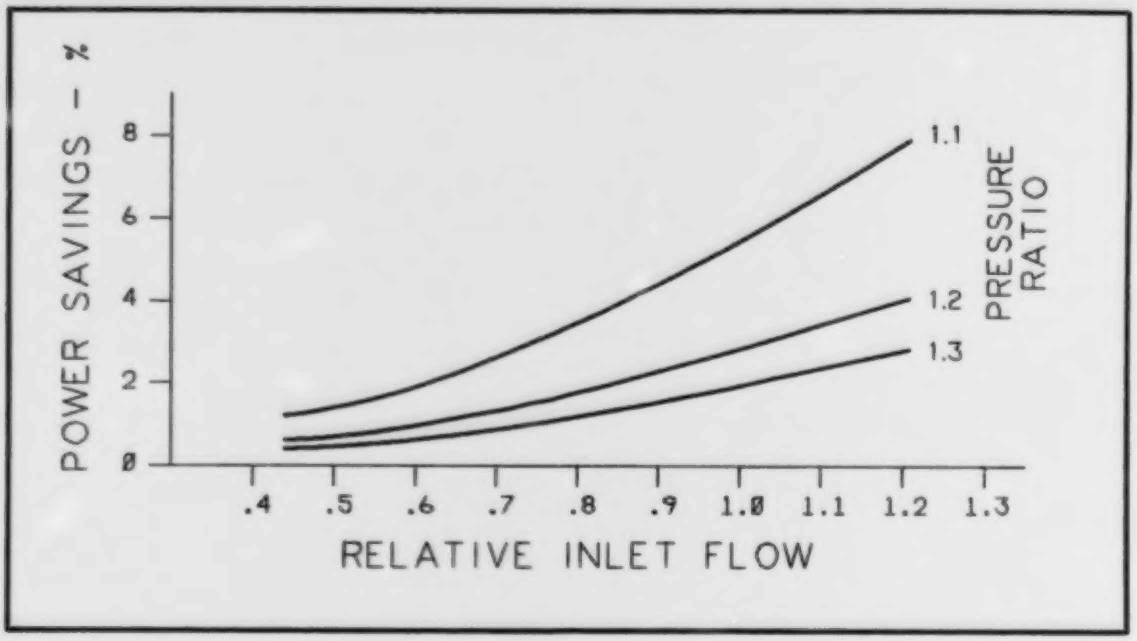


FIGURE 7 EFFECTIVENESS OF AXIAL INLET CONVERSION ON CONVENTIONAL PIPELINE BOOSTERS WITH SIDE INLET

CONCLUSIONS

The following conclusions can be drawn of the effectiveness of converting a conventional pipeline booster to axial inlet:

1. The viability of this concept has proven itself after nearly two years of service.
2. Power savings of three to eight percent have been realized on units converted to date from the compressor conversion alone. Higher gains are possible.
3. The magnitude of savings depends on relative volume flow and pressure ratio requirements.
4. Conversion to axial inlet does not supersede the need for aerodynamic reconfiguration as operating conditions change.
5. Some maintenance limitations result from this conversion.

6. While conversion to axial inlet has proven to be very cost effective, further performance gains could be realized from the installation of a true axial inlet pipeline booster.

REFERENCES

1. Meier, R. H., Schiller, R. N.: "Development and Testing of a New High Flow Centrifugal Pipeline Booster", Sixth Turbomachinery Symposium (1977), Texas A&M University, Houston, Texas.
2. Meier, R. H., Rhea, C. S.: "Centrifugal Compressor Testing Experience and Practice", ASME Paper No. 82-GT-320.

ACKNOWLEDGEMENT

The author wishes to thank TransCanada Pipelines, Ltd. for their interest in this program, which has been instrumental in its realization. TCPL's assistance with data and photographs and their permission to present this paper are also greatfully recognized.

Printed in USA

THE DEVELOPMENT OF A COMPRESSOR STATION HEALTH MONITORING SYSTEM

T. Van Hardeveld, P.Eng.
Maintenance Engineering Specialist
Nova, An Alberta Corporation
P.O. Box 2330, 9888 Jasper Ave
Edmonton, Alberta, T5J 2R1

ABSTRACT

Over the past few years, there has been a renewed interest in predictive maintenance techniques, including health or condition monitoring. Economic pressures have certainly contributed to the desire to operate equipment more efficiently and with lower maintenance cost but the main impetus to the development of better health monitoring techniques has been the availability of low-cost and powerful microcomputers and enhanced software development tools. These factors have led NOVA, An Alberta Corporation to develop a condition monitoring system, called the Compressor Health Monitoring System.

The following describes the development of this system for NOVA's compressor station facilities, a process that has taken place over the past 20 months. In that time frame, the system has been prototyped, tested, enhanced, and implemented at 14 compressor sites. More than 3 years of accumulated experience has been gained and a number of benefits have already been identified where problems have been noted and diagnosed successfully. Most importantly, operator acceptance has been very high since the system has served to enhance their role as the first line of defence, a function that had been somewhat eroded by equipment automation.

The objectives that one needs to consider when setting up a health monitoring system are presented first. These are followed by a description of the major system functions. The development strategy is then outlined because of its importance in achieving the final objectives. Finally, some of the benefits are introduced.

1.0 INTRODUCTION

The Compressor Health Monitoring System is a condition monitoring system for compressor stations that can be used by operating staff on-site and by centralized Maintenance Planning personnel to facilitate the prediction of deterioration in machinery condition, performance, and efficiency. It replaces a well-established procedure of compressor unit and auxiliary log sheets and other paper-based forms to record and disseminate compressor station information. The prime objective is to provide a complete information environment for the operator with respect to equipment operating data. However, analysis of this data for maintenance purposes also offers many benefits and further enhancement of the system will take place, as more experience is gained, to implement analysis routines for Maintenance Planning in addition to those already in place.

To achieve these objectives, data is gathered manually by an operator using a hand-held data collector and is then uploaded to a microcomputer on-site. The data is processed and presented to the operator in a friendly fashion for interpretation and analysis to aid in his function as the first line of defense. The data is then transmitted to Maintenance Planning for use in planning maintenance activities.

The system initially is intended to be stand-alone with respect to the on-site instrumentation since most compressor stations, especially the older ones, are not adequately equipped to allow direct acquisition of data. However, in the future, direct acquisition is expected to become the normal manner in which most of the data is obtained. In the meantime, the stand-alone system will provide good capability for medium to long term trending; at a later date, direct acquisition will add the ability to do short-term trending and alarm monitoring.

The system is intended to address all aspects of compressor station monitoring, including all unit operating parameters, consumables, and data such as running hours. In contrast, the emphasis in many health monitoring systems available commercially is on vibration techniques more than performance, efficiency, or the more mundane monitoring of auxiliary systems. A sample of other approaches is given in references 1-5 at the end of the paper. The NOVA system includes gas turbine performance monitoring techniques that consist of the calculation of deviation from tested baselines in order to detect gas path deterioration (references 6-7). This concept is not new but it has been applied in only a few cases, likely because of the cost (references 8-9).

2.0 OBJECTIVES OF HEALTH MONITORING

2.1 DATA GATHERING

The first objective of the system is to collect accurate and consistent data and to detect possible errors at the source. This is done by incorporating edit checks within the hand-held data collector and the computer and by rejecting improper readings.

Another aspect of data gathering is the fact that the flow of information to Maintenance Planning is accelerated by the ability to send data over communication lines on as frequent a basis as necessary.

2.2 PROCESSING OF DATA

The second purpose of the system is to enable processing of data to enhance its usefulness. This is done, first of all, by allowing alert checks to be performed on any operating parameter (original or calculated) so that abnormal operating conditions can be automatically flagged for the operator.

Secondly, processing of data includes calculations of performance and efficiency since these are invariably affected by equipment deterioration and point to increasing cost. For a gas turbine, deterioration can also be detected by comparing normalized readings against a tested baseline.

2.3 INTERPRETATION OF DATA

Manual methods of handling operating data, such as log sheets, often fail because there is no practical way of presenting data so that it can be interpreted. A powerful method of showing data is to use graphics. One way is to trend operating data against unit operating hours with the flexibility to plot several parameters together to allow comparison. An added advantage is that the amount of data scatter will show whether data is consistent.

Operating conditions at a point in time can be graphed easily. One example is the centrifugal compressor wheel curve which will inform the operator where the operating point is relative to surge, choke, head across the unit, and areas of high or low efficiency.

2.4 ANALYSIS OF DATA

The amount of operating data, even when taken manually, increases beyond the ability of people to analyze it unless computerized methods are used. Operators can benefit by being able to review past operating conditions or problems. Maintenance Planning faces an even more difficult task in trying to evaluate long-term trends and to apply these to improving the effectiveness of preventive maintenance.

3.0 DEVELOPMENT STRATEGY

3.1 INTRODUCTION

It was recognized at the beginning of the project that a development strategy was needed to increase the chance of success. The basic idea of health monitoring seemed sound enough to anyone with operating or maintenance experience but its implementation was not going to be easy, largely because of two factors. The first one was that it was going to affect a large and diverse group of field staff, many of whom were not trained to work with microcomputers. The second factor was that it had to work for many different types of compressor station equipment while at the same time maintaining a measure of consistency. The fact that no software existed that would meet NOVA's requirements, dictated that the software be developed internally.

3.2 INVOLVEMENT OF OPERATING STAFF

The core team consisted of four representatives from the operating areas, both operators and control technicians, a computer specialist, and an engineer from the Technical Services department as project leader. This team developed the design requirements after visiting and discussing the objectives with about ten compressor sites. The field reps have remained involved in testing and implementation of the first prototypes and have also been responsible for the installation of the software in their districts. Their contribution has been a major reason for the success of the project.

3.3 DESIGN REQUIREMENTS

The important question in specifying the design requirements was how far to go in meeting the objectives. After due consideration of the relative benefits, it was decided that the best course of action was to develop, as a first step, a basic stand-alone system with respect to the acquisition of data. A hand-held data collector would be added once the prototypes showed the feasibility of the project. The system could then be implemented at minimum cost at compressor sites with many benefits already being obtainable; the same software would also be usable by Maintenance Planning. Specific Maintenance Planning requirements were likewise deferred until experience could be gained. The second step, that of direct acquisition of data from on-site instrumentation, would lag somewhat but would be developed largely for new compressor stations and those where controls upgrades had occurred. A prototype of a direct acquisition system is currently being developed for testing later this year.

3.4 SOFTWARE DEVELOPMENT

The approach to software development constituted a major decision point. It was decided that the most appropriate development methodology would be to prototype the two predominant types of compressors, that is, a gas turbine/centrifugal compressor and a reciprocating engine and compressor. The results of this testing would then allow enhancement based on actual experience, thereby also improving user acceptance. This approach has turned out to be the correct one and will certainly reduce drastically the need to upgrade the software in the future.

The second significant aspect of software development was the requirement to produce a system that could be reconfigured easily for all types of compressor equipment. For this purpose, most variable data was set up in a database which can be easily changed;

as a result, a new application can be set up in about a week.

Another prime objective in software design was to produce a system that was easy to use and to learn for the operator; this was more than adequately achieved with an operator needing about 2-3 hours to become familiar with all of the functions.

3.5 HARDWARE CONSIDERATIONS

The hardware can be broken into three components: the data collector, a microcomputer on-site, and a similar computer at a centralized location for Maintenance Planning.

The hand-held data collector is currently under development by NOVA-Husky Research Corporation. Its main feature is that it will be intrinsically safe and will have full CSA approval for use in hazardous areas. The program will be contained in the data collector and site-specific data will be downloaded from the microcomputer before a log is taken. The operator selects the appropriate log and enters data in a pre-determined sequence; edit and alert checks are carried out for each reading. The data is then uploaded into the microcomputer.

The main hardware on-site is an IBM PC-AT or equivalent with a hard disk and modem for communications; NOVA is using a DEC VaxMate (equivalent to the IBM PC-AT) because it networks easily with existing NOVA computer equipment. The use of generic hardware allowed for the widest choice of software development tools. The computer used at the centralized location is the same as that at the compressor stations except that a larger hard disk is used.

3.6 INSTRUMENTATION

The final aspect of development strategy was to ensure that the critical data points be measured with sufficient accuracy to achieve good results. In particular, monitoring instrumentation, such items as speeds, compressor discharge pressure (engine), exhaust gas temperature, gas compressor temperatures and pressures was checked for accuracy and was changed out if necessary; barometric pressure indicators were supplied to all stations. The calculation of baseline deviation pointed out a number of instances of poor instrument calibration.

4.0 MAJOR FUNCTIONS

4.1 UNIT AND AUXILIARY LOGS

The logging function constitutes the core of the system. It closely parallels the manual log sheets that are widely used in recording compressor unit and auxiliary operating data. However, some crucial improvements have been made. The most important one is the hand-held data collector which eliminates paper. Accuracy is improved because edit checks can be done at the point of data collection. At the same time, an alert check is done and the operator is notified immediately of abnormal conditions; they are referred to as 'alerts' so as not to confuse them with 'alarms' which are produced by the unit control system. As backup or alternative to the data collector, a blank log sheet can be printed off and readings can be entered manually via the keyboard of the microcomputer.

After log data is entered, either by the data collector or by the keyboard, calculations are performed. These normally consist of power output, fuel efficiency, gas flow, compressor head, and compressor efficiency. In addition, gas turbine parameters, such as speed, compressor discharge pressure, and exhaust gas temperature, are normalized so that deviations from baselines can be determined. The inclusion of calculated parameters in the log provides important benefits in interpreting equipment health. Finally, a summary of alarms is presented and the operator can record applicable comments.

Each log can be viewed afterwards as shown in figure 1. The operating parameters are organized into logical groups, such as engine operating conditions, compressor operating conditions, and the lube oil system. Alerts, along with the log comments, are also accessible on a separate alert summary which contains this information for all logs at the station (see figure 2). A daily report with all logs taken on a specific day can be printed if desired.

Logs are also usually set up for auxiliaries such as auxiliary power units, air compressors, and heating systems.

4.2 OPERATING GRAPHS

Some log data is better shown on a graph. The major example is the gas compressor wheel curve of which an example is given in figure 3. Since certain abnormal conditions are affected by the compressor operating point, this graph can aid considerably in understanding problems. For instance, vibration will increase if the compressor is operating close to surge or in choke without an abnormal condition existing. As well, the wheel curve will show if the compressor is operating in an efficient area.

Other examples of operating curves are engine performance baselines, such as compressor discharge pressure as a function of speed, and vibration baselines, that is, vibration against engine speed. An example of a vibration baseline is shown in figure 4.

4.3 TREND GRAPHS

A trend graph can be produced for any operating parameter on a log. The trend is against total operating hours on the unit so that time when the unit is inactive is ignored. The trend graph is probably the most valuable method available for interpreting data, detecting deterioration, and analyzing problems.

Any of the log data can be displayed on a trend graph and up to four lines can be shown simultaneously. The graph initially depicts 1500 operating hours of data (or two months of continuous running); one can then zoom to 150 hours (one week) and 24 hours (one day). There is a cursor which can be moved from point to point with the values of the data being displayed on the right-hand side of the graph. This allows the operator to determine the exact log for, as an example, a high reading. The alert lines are also shown individually and, for multiple lines, the operator can toggle between them. An example of a trend graph is given in figure 5.

4.4 DAILY OPERATING DATA

For compressor units, it is important to monitor the status of the unit, that is, what amount of time it is running, on no demand, or undergoing maintenance, either scheduled or unscheduled. This data is used to keep track of total running hours on the unit and by Maintenance Planning to calculate availability and reliability. Figure 6 gives an example.

For auxiliaries, only a daily record of total operating hours is necessary as shown in figure 7. A daily average usage is calculated so that, for air compressors in particular, excessive air loss and air compressor efficiency can be monitored.

4.5 BACKGROUND INFORMATION

Background information refers to the equation constants that may change from time to time. It includes the constants for gas composition, the main gas and fuel gas flow devices, base temperature and pressure, the gas turbine performance baselines, and others. As well, the constants for the wheel curve are included in order to calculate deviations from the tested or manufacturer's curves so that impeller erosion can be detected. Up to five wheel curves can be accommodated in case different compressor configurations are used. Figure 8 shows a sample screen for the background information.

4.6 CONSUMABLES

An important function of the operator is to monitor the fluid levels of consumables and to replace fluids where necessary. For this purpose, the system contains a section on consumables which allows the recording of fluid additions, either normal, abnormal, or replacement, and the calculation of usage and hours since last replacement (see figure 9). The determination of fluid usage applies only to normal addition and is done by dividing the number of operating hours since the last normal addition into the amount of fluid added. The usage can be shown on a trend graph.

4.7 SYSTEM FUNCTIONS

System functions consist of various housekeeping items. First, there exists a function to do regular backups of all of the data onto a diskette (or multiple diskettes) in case of hardware problems; there is a reverse procedure to restore the data. Secondly, on-site staff can archive data to off-load the hard disk; archiving entails a complete backup of data, so that it can be restored later if needed, and then a selective deleting of information. Finally, there is a function to allow the transmission of data via modem to a centralized location.

5.0 BENEFITS

In summary, the following benefits have accrued or are anticipated from the Compressor Health Monitoring System:

1. Operating data is more accurate because of edit checks and is more complete because of the inclusion of calculated values. It has been found that the operators are more conscious of data accuracy because they know it will be used for analysis.

2. Operators have a better knowledge of unit operating conditions through the graphing of the compressor wheel curve, the vibration baselines, etc. and can better evaluate their effect on abnormal conditions.

3. On-site staff are alerted immediately of abnormal conditions, when an alert value is exceeded, and can more easily determine the cause by comparing the appropriate operating parameters on a trend graph. The setting of alert values when the system was first implemented was an interesting process since it became clear that the operators had incomplete and conflicting understandings of what the actual levels should be.

4. By checking trends periodically, an operator or Maintenance Planning technician can detect equipment deterioration and initiate action before a failure. This can save downtime and maintenance costs and aids in the diagnosis of problems. For example, a sudden and drastic increase in exhaust gas temperature spread on a 3000 HP reciprocating engine and compressor was traced to a change in air/fuel ratio caused by a drop in ambient temperature; a simple adjustment of the air/fuel ratio eliminated the problem. Previously, air/fuel ratio was not calculated by the operator and, as has happened in the past, major maintenance would have been carried out to try to find the problem. This problem also pointed out that the air/fuel ratio control was not adequate for the complete range of ambient conditions and more frequent adjustment is required.

5. The calculation of fuel consumption and other indicators of efficiency will save on operating costs although these are difficult to estimate. In the past, deterioration in engine compressor efficiency has not been detected since deviation from a tested baseline was not computed except at formal performance tests. As well, impeller erosion in the gas compressor has gone undetected for a long time. In both of these cases, considerable savings can be obtained although none have been noted so far.

6. The improvement in the accuracy and accessibility of information for analysis has important implications for the maintenance planning process. The ability to monitor deterioration strengthens preventive maintenance because the scope and frequency of inspections can be modified by actual operating data. As a result, inspections can be done less often if the equipment is running well or more often if it deteriorates more rapidly than expected.

REFERENCES

1. Mitchell, John S., How To Develop a Machinery Monitoring Program, Sound and Vibration, February 1984.

2. Cummings, J. M., Guidelines for Machinery Data Logging, Hydrocarbon Processing, January 1985.

3. Ricca, P.M., and Bradshaw, P.M., How trans-Alaska pipeline system's rotating maintenance program evolved, Oil & Gas Journal, Aug 13, 1984.

4. Jorgensen, Paul A., Compressor Station Data Acquisition and Diagnostics, The American Gas Association Distribution/Transmission Conference, Chicago, Illinois, April 28, 1986.

5. Scott, J.N., Reducing Turbo Machinery Operating Costs with Regular Performance Testing, The International ASME Gas Turbine Conference and Exhibit, June 8-12, 1986.

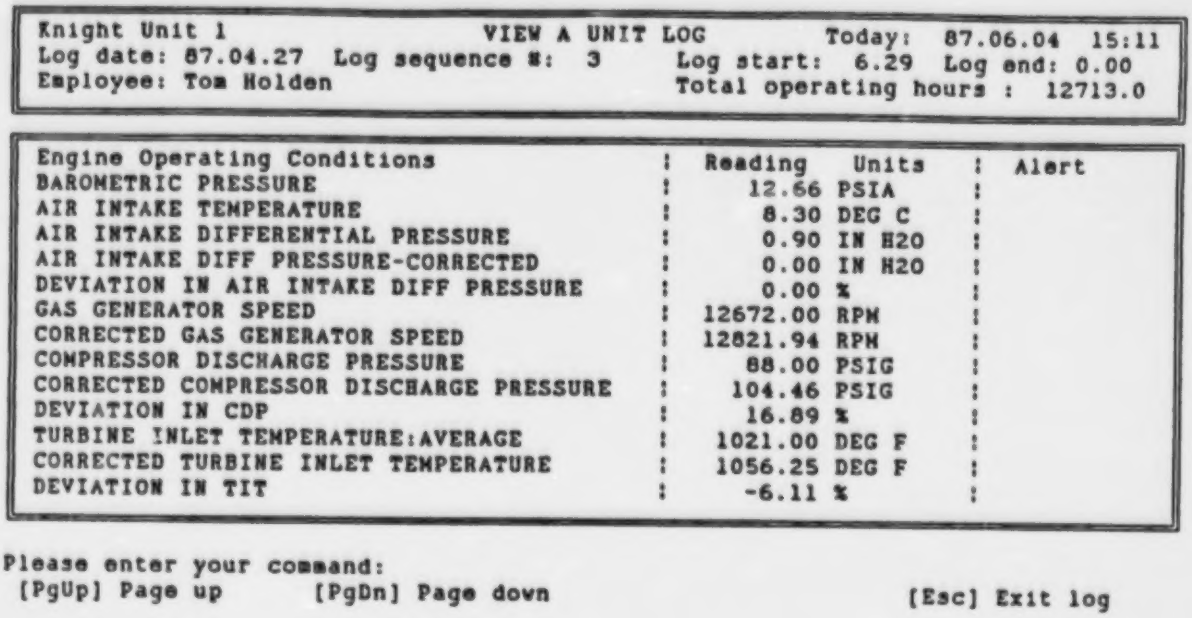
6. Urban, L.A., Condition Monitoring of Turbine Engines and Gas Compressors for Gas Line Pumping, Gas Turbine Operations and Maintenance Symposium, Edmonton, Alberta, October 21-23, 1974.

7. Dupuis, R.J., Saravanamuttoo, H.I.H., and Rudnitski, D.M., The Use of Fault Matrices in On-Condition Monitoring, Sixth NRCC Symposium on Gas Turbine Operations and Maintenance, Montreal, September 1985.

8. Joshi, Kam, Experience with Gas Turbine Performance Monitoring Systems, Second NRCC Symposium on Gas Turbine Operation and Maintenance, June 1977.

9. Fairbairn, G.W., A British Gas Corporation on-line engine condition monitoring pilot scheme, Seminar organized by the Fluid Machinery Committee of the Power Industries Division of the Institution of Mechanical Engineers, England, June 6, 1985.

FIGURE 1 - Compressor Unit Log

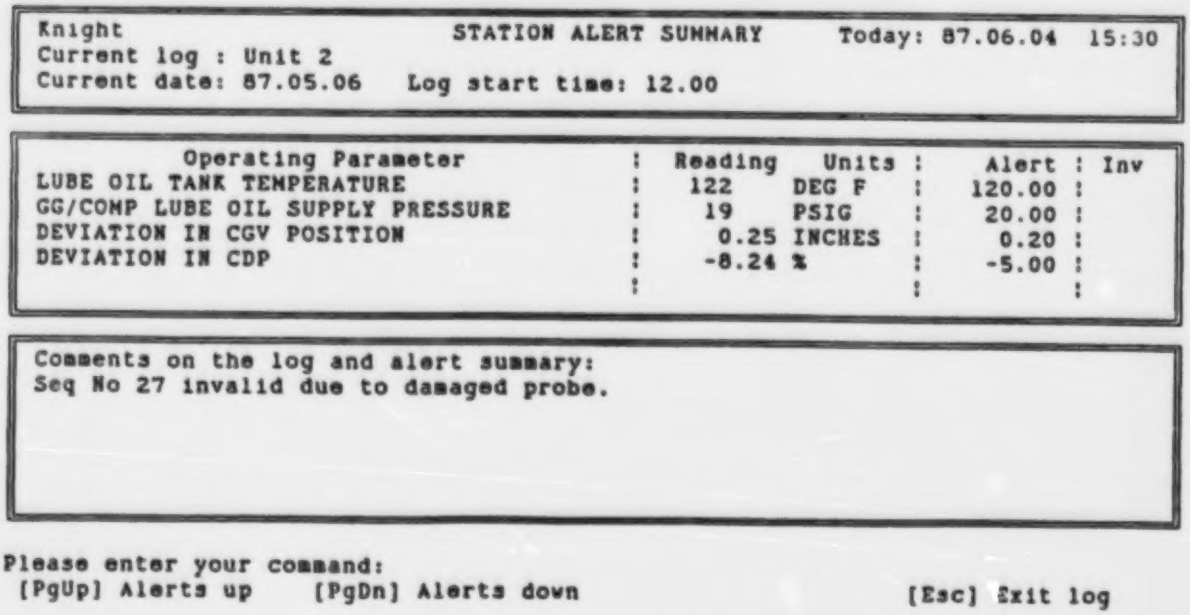


```
Knight Unit 1                VIEW A UNIT LOG        Today:  87.06.04  15:11
Log date: 87.04.27  Log sequence #:  3     Log start:  6.29  Log end: 0.00
Employee: Tom Holden                       Total operating hours :  12713.0

Engine Operating Conditions                 !  Reading   Units   !  Alert
BAROMETRIC PRESSURE                         !     12.66 PSIA     !
AIR INTAKE TEMPERATURE                      !      8.30 DEG C    !
AIR INTAKE DIFFERENTIAL PRESSURE            !      0.90 IN H2O   !
AIR INTAKE DIFF PRESSURE-CORRECTED          !      0.00 IN H2O   !
DEVIATION IN AIR INTAKE DIFF PRESSURE       !      0.00 %        !
GAS GENERATOR SPEED                         !  12672.00 RPM      !
CORRECTED GAS GENERATOR SPEED               !  12821.94 RPM      !
COMPRESSOR DISCHARGE PRESSURE               !     88.00 PSIG     !
CORRECTED COMPRESSOR DISCHARGE PRESSURE     !    104.46 PSIG     !
DEVIATION IN CDP                            !     16.89 %        !
TURBINE INLET TEMPERATURE:AVERAGE           !   1021.00 DEG F    !
CORRECTED TURBINE INLET TEMPERATURE         !   1056.25 DEG F    !
DEVIATION IN TIT                            !     -6.11 %        !

Please enter your command:
 [PgUp] Page up      [PgDn] Page down                       [Esc] Exit log
```

FIGURE 2 - Station Alert Summary



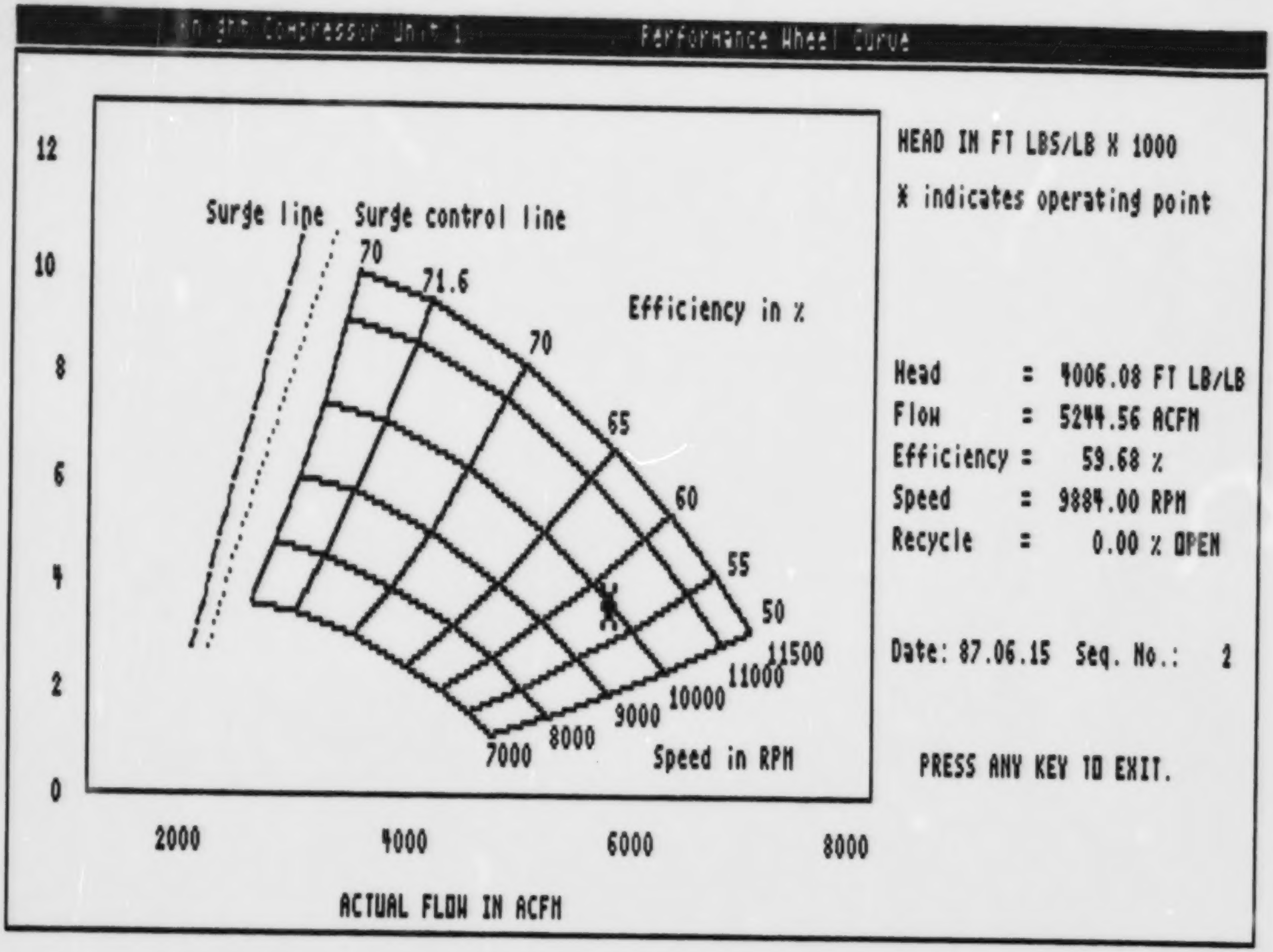
```
Knight                         STATION ALERT SUMMARY     Today: 87.06.04  15:30
Current log : Unit 2
Current date: 87.05.06   Log start time: 12.00

          Operating Parameter           !  Reading   Units  !    Alert ! Inv
LUBE OIL TANK TEMPERATURE               !   122     DEG F   !   120.00 !
GG/COMP LUBE OIL SUPPLY PRESSURE        !    19     PSIG    !    20.00 !
DEVIATION IN CGV POSITION               !    0.25 INCHES    !     0.20 !
DEVIATION IN CDP                        !   -8.24 %         !    -5.00 !
                                        !                   !          !

Comments on the log and alert summary:
Seq No 27 invalid due to damaged probe.

Please enter your command:
 [PgUp] Alerts up    [PgDn] Alerts down                     [Esc] Exit log
```

FIGURE 3 - Compressor Wheel Curve


Knight Compressor Unit 1
Performance Wheel Curve
12
10
8
6
4
2
0
Surge line
Surge control line
70
71.6
70
65
60
55
50
Efficiency in %
7000
8000
9000
10000
11000
11500
Speed in RPM
2000
4000
6000
8000
ACTUAL FLOW IN ACFM

HEAD IN FT LBS/LB X 1000

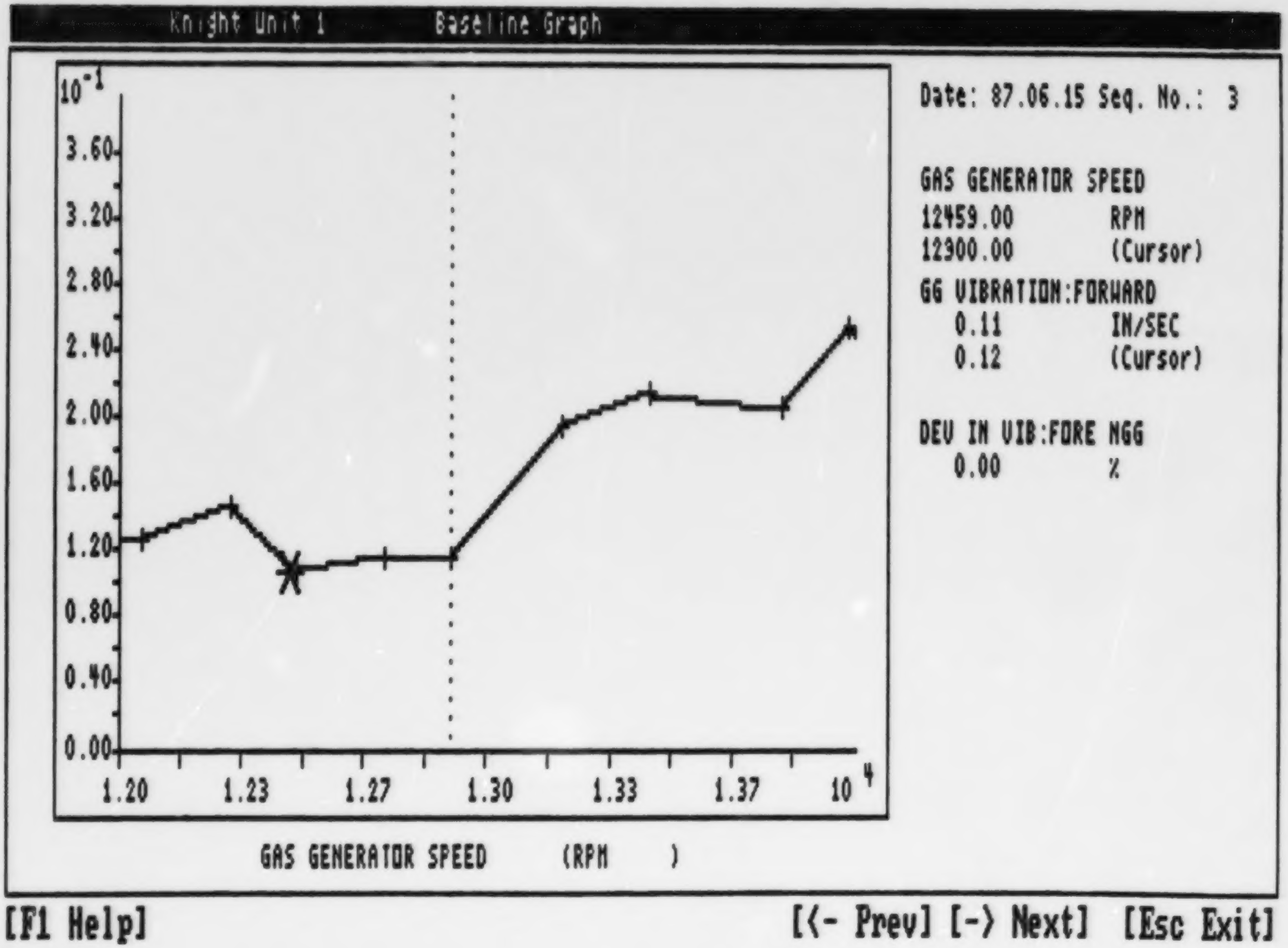
X indicates operating point

```
Head       =  4006.08 FT LB/LB
Flow       =  5244.56 ACFM
Efficiency =    59.68 %
Speed      =  9884.00 RPM
Recycle    =     0.00 % OPEN

Date: 87.06.15  Seq. No.:   2

PRESS ANY KEY TO EXIT.
```

FIGURE 4 - Vibration Baseline Graph

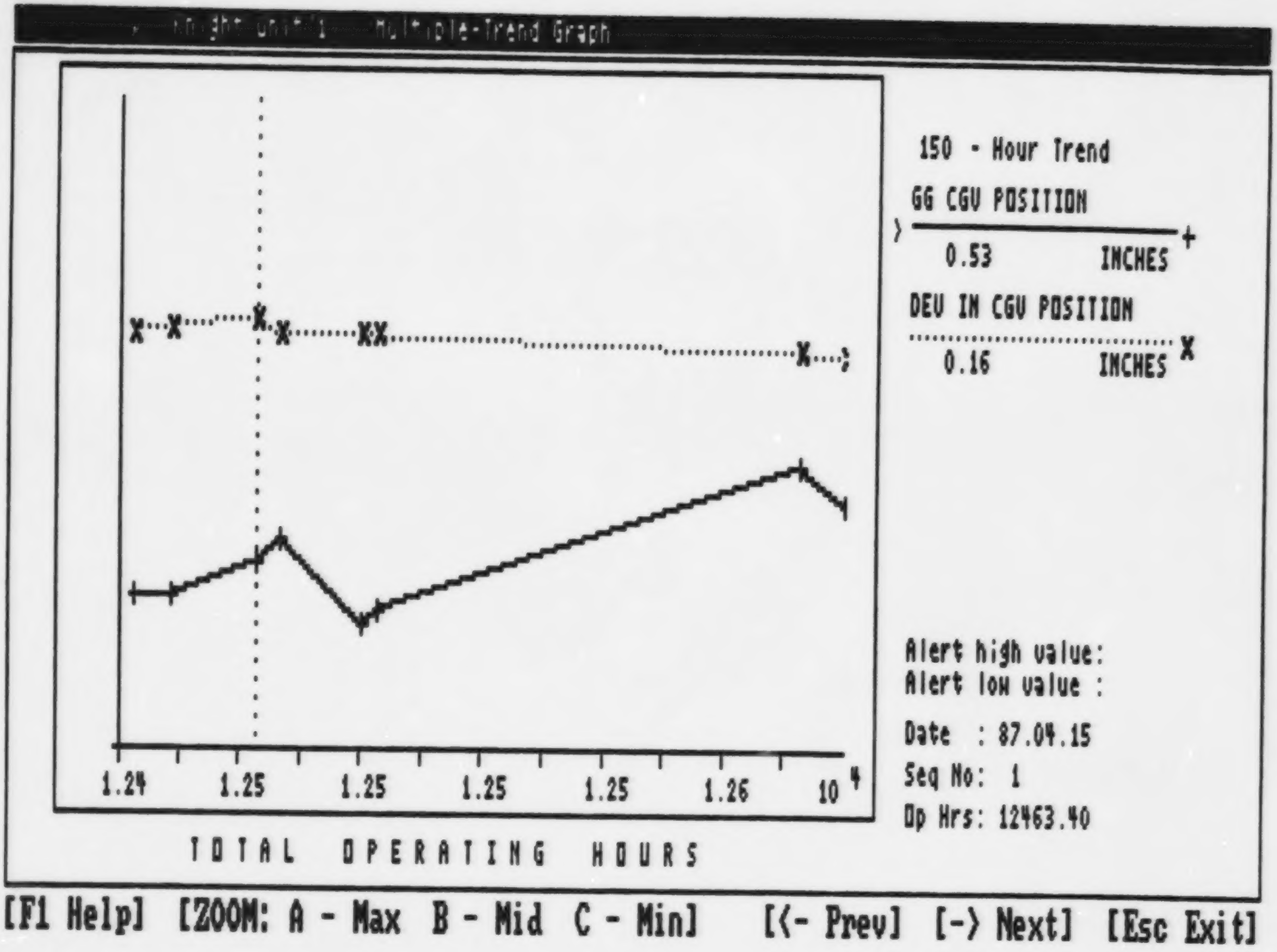

Knight Unit 1
Baseline Graph
10^-1
3.60
3.20
2.80
2.40
2.00
1.60
1.20
0.80
0.40
0.00
1.20
1.23
1.27
1.30
1.33
1.37
10^4
GAS GENERATOR SPEED (RPM)
[F1 Help]
[<- Prev] [-> Next] [Esc Exit]

```
Date: 87.06.15 Seq. No.:  3

GAS GENERATOR SPEED
12459.00        RPM
12300.00        (Cursor)
GG VIBRATION:FORWARD
   0.11         IN/SEC
   0.12         (Cursor)

DEV IN VIB:FORE NGG
   0.00         %
```

FIGURE 5 - Trend Graph



Knight unit 1 — Multiple-Trend Graph

150 - Hour Trend

GG CGV POSITION
0.53 INCHES

DEV IN CGV POSITION
0.16 INCHES

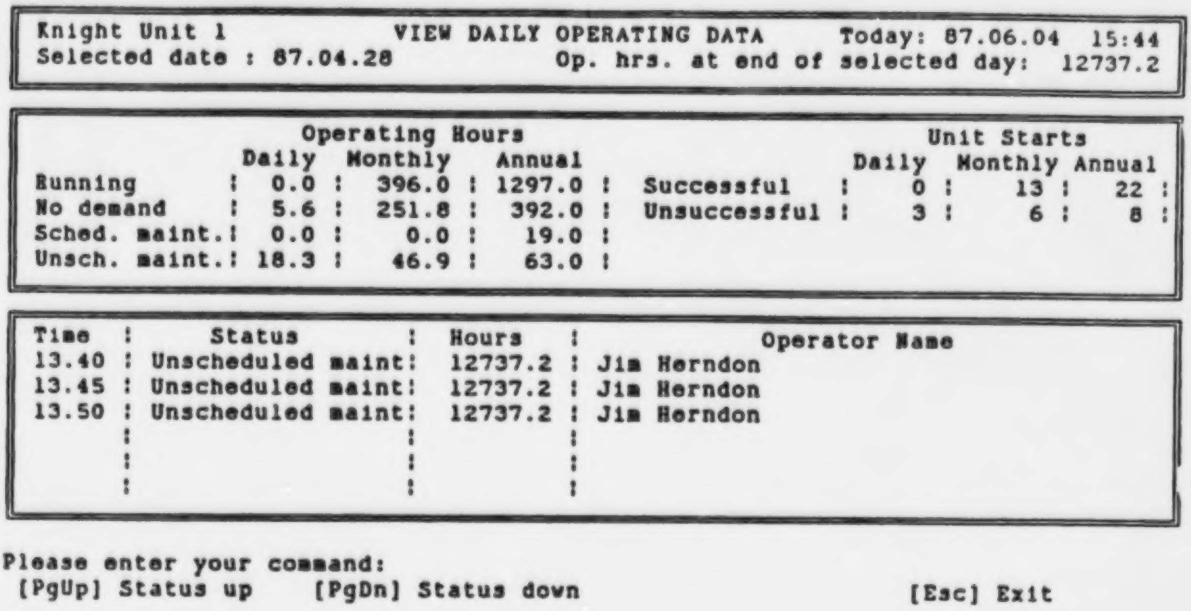
Alert high value:
Alert low value :

Date : 87.04.15
Seq No: 1
Op Hrs: 12463.40

[F1 Help] [ZOOM: A - Max B - Mid C - Min] [<- Prev] [-> Next] [Esc Exit]

1.24
1.25
1.25
1.25
1.25
1.26
10 4
TOTAL OPERATING HOURS

FIGURE 6 - Daily Operating Data



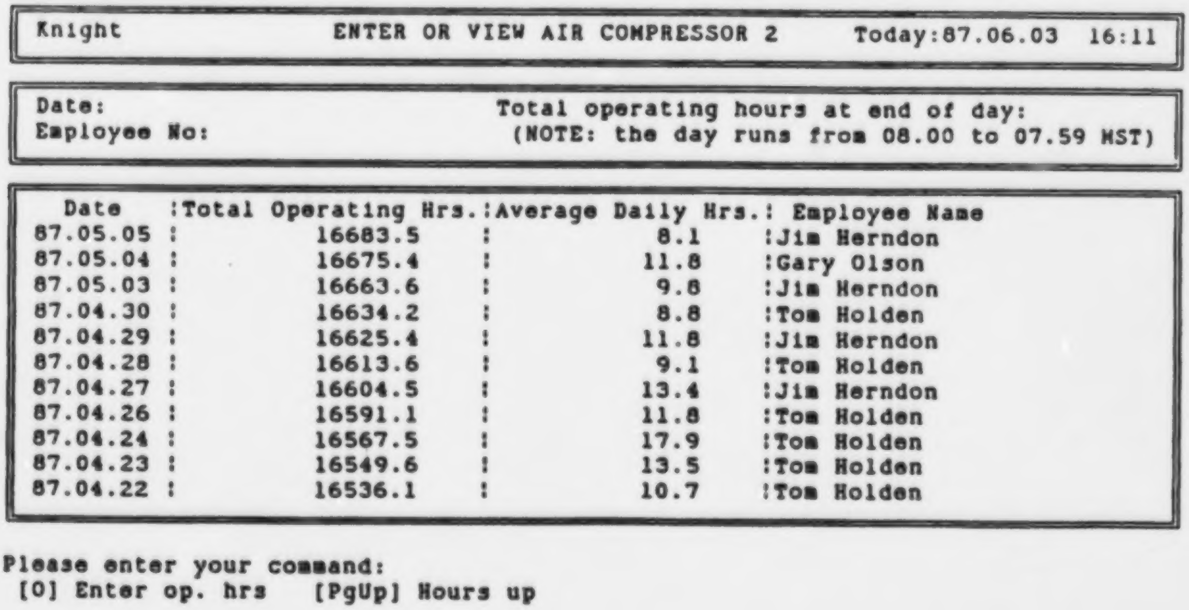
```
Knight Unit 1            VIEW DAILY OPERATING DATA     Today: 87.06.04  15:44
Selected date : 87.04.28          Op. hrs. at end of selected day:  12737.2

                   Operating Hours                              Unit Starts
               Daily  Monthly   Annual                     Daily  Monthly Annual
Running      ¦  0.0 ¦  396.0 ¦ 1297.0 ¦  Successful   ¦     0 ¦      13 ¦    22 ¦
No demand    ¦  5.6 ¦  251.8 ¦  392.0 ¦  Unsuccessful ¦     3 ¦       6 ¦     8 ¦
Sched. maint.¦  0.0 ¦    0.0 ¦   19.0 ¦
Unsch. maint.¦ 18.3 ¦   46.9 ¦   63.0 ¦

Time  ¦      Status       ¦  Hours    ¦               Operator Name
13.40 ¦ Unscheduled maint¦  12737.2 ¦ Jim Herndon
13.45 ¦ Unscheduled maint¦  12737.2 ¦ Jim Herndon
13.50 ¦ Unscheduled maint¦  12737.2 ¦ Jim Herndon
      ¦                   ¦          ¦
      ¦                   ¦          ¦
      ¦                   ¦          ¦

Please enter your command:
 [PgUp] Status up    [PgDn] Status down                         [Esc] Exit
```

FIGURE 7 - Auxiliary Operating Data



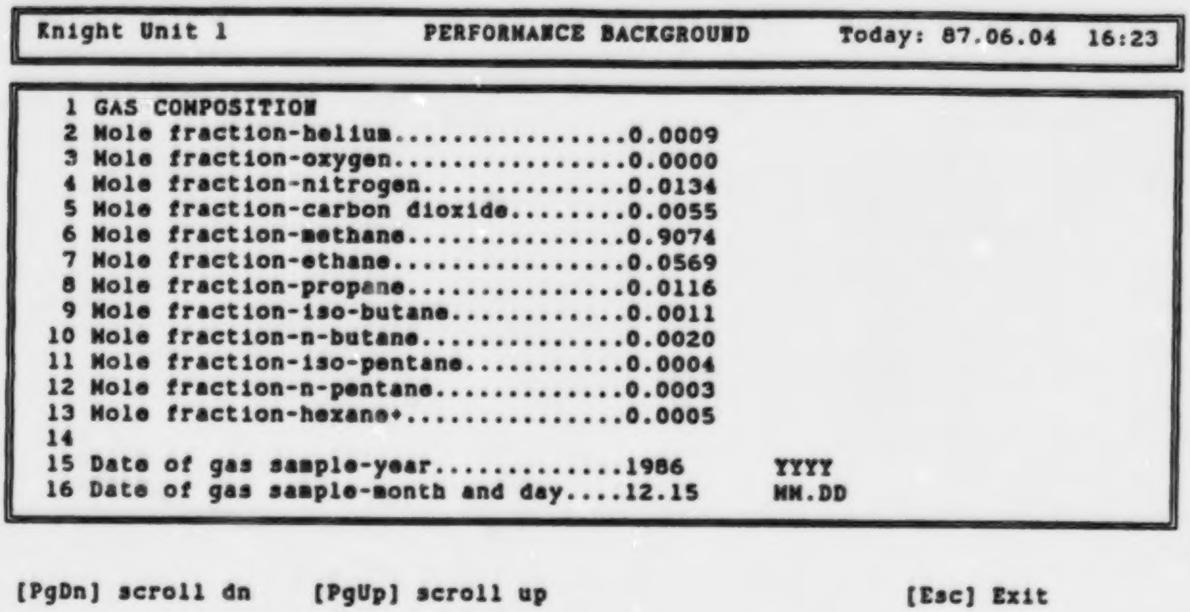
```
Knight               ENTER OR VIEW AIR COMPRESSOR 2     Today:87.06.03  16:11

Date:                                Total operating hours at end of day:
Employee No:                          (NOTE: the day runs from 08.00 to 07.59 MST)

  Date    ¦Total Operating Hrs.¦Average Daily Hrs.¦ Employee Name
87.05.05 ¦           16683.5    ¦               8.1   ¦Jim Herndon
87.05.04 ¦           16675.4    ¦              11.8   ¦Gary Olson
87.05.03 ¦           16663.6    ¦               9.8   ¦Jim Herndon
87.04.30 ¦           16634.2    ¦               8.8   ¦Tom Holden
87.04.29 ¦           16625.4    ¦              11.8   ¦Jim Herndon
87.04.28 ¦           16613.6    ¦               9.1   ¦Tom Holden
87.04.27 ¦           16604.5    ¦              13.4   ¦Jim Herndon
87.04.26 ¦           16591.1    ¦              11.8   ¦Tom Holden
87.04.24 ¦           16567.5    ¦              17.9   ¦Tom Holden
87.04.23 ¦           16549.6    ¦              13.5   ¦Tom Holden
87.04.22 ¦           16536.1    ¦              10.7   ¦Tom Holden

Please enter your command:
 [O] Enter op. hrs   [PgUp] Hours up
```

FIGURE 8 - Background Information

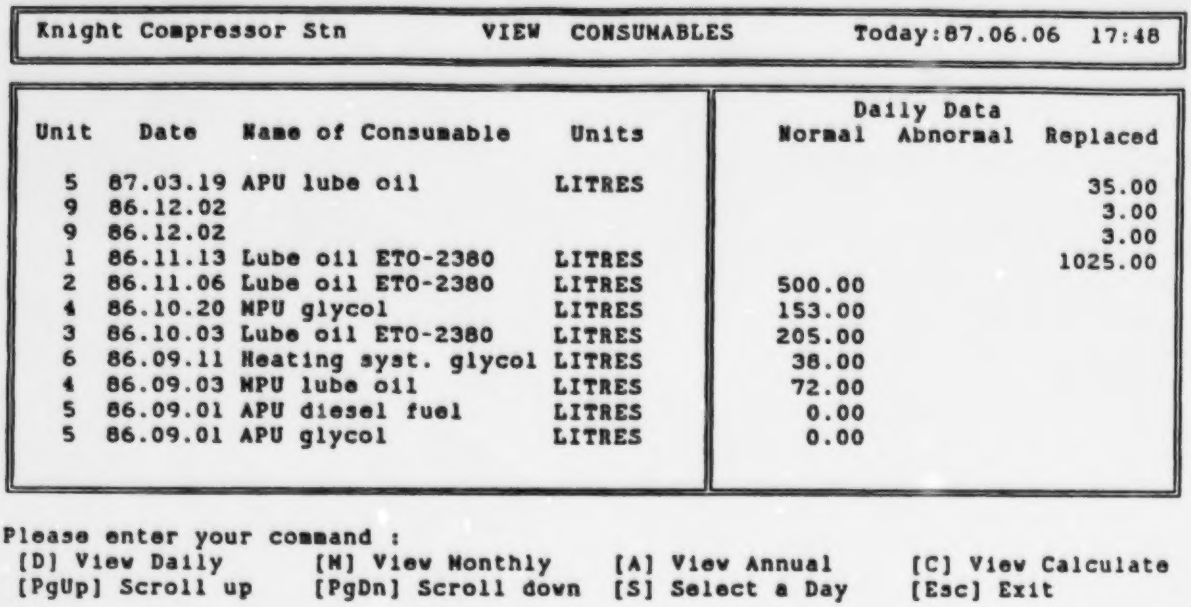


```
 Knight Unit 1                PERFORMANCE BACKGROUND        Today: 87.06.04  16:23

   1 GAS COMPOSITION
   2 Mole fraction-helium.................0.0009
   3 Mole fraction-oxygen.................0.0000
   4 Mole fraction-nitrogen...............0.0134
   5 Mole fraction-carbon dioxide.........0.0055
   6 Mole fraction-methane................0.9074
   7 Mole fraction-ethane.................0.0569
   8 Mole fraction-propane................0.0116
   9 Mole fraction-iso-butane.............0.0011
  10 Mole fraction-n-butane...............0.0020
  11 Mole fraction-iso-pentane............0.0004
  12 Mole fraction-n-pentane..............0.0003
  13 Mole fraction-hexane+................0.0005
  14
  15 Date of gas sample-year.............1986      YYYY
  16 Date of gas sample-month and day....12.15     MM.DD

[PgDn] scroll dn    [PgUp] scroll up                          [Esc] Exit
```

FIGURE 9 - Consumables

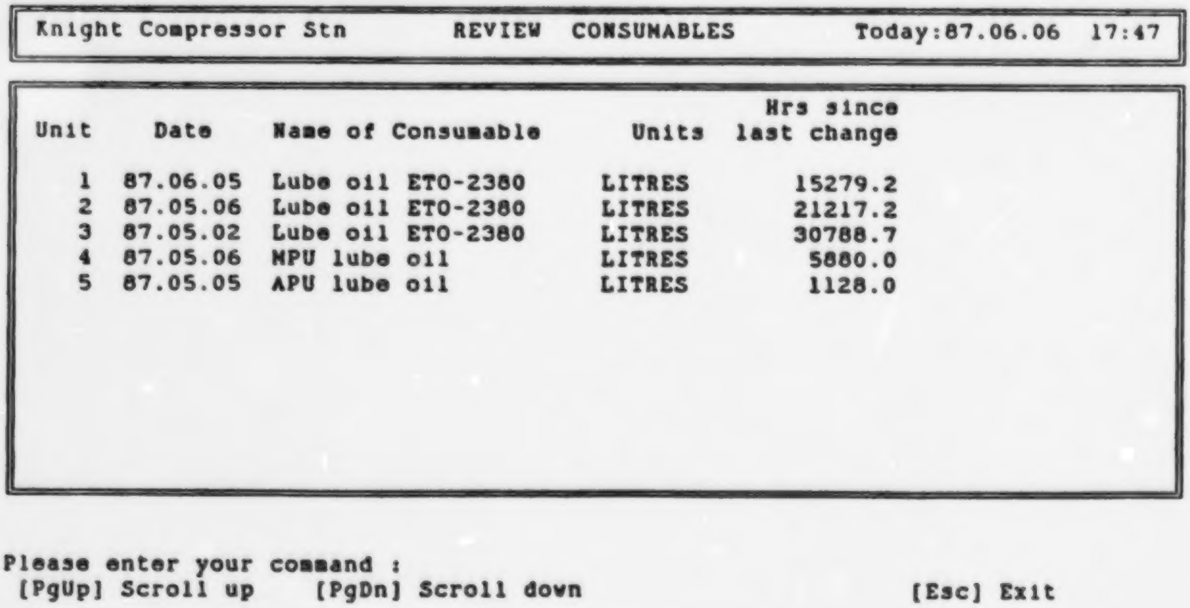


```
Knight Compressor Stn          VIEW  CONSUMABLES          Today:87.06.06  17:48

                                                           Daily Data
Unit   Date    Name of Consumable     Units       Normal  Abnormal  Replaced

  5  87.03.19 APU lube oil           LITRES                             35.00
  9  86.12.02                                                            3.00
  9  86.12.02                                                            3.00
  1  86.11.13 Lube oil ETO-2380      LITRES                           1025.00
  2  86.11.06 Lube oil ETO-2380      LITRES       500.00
  4  86.10.20 MPU glycol             LITRES       153.00
  3  86.10.03 Lube oil ETO-2380      LITRES       205.00
  6  86.09.11 Heating syst. glycol   LITRES        38.00
  4  86.09.03 MPU lube oil           LITRES        72.00
  5  86.09.01 APU diesel fuel        LITRES         0.00
  5  86.09.01 APU glycol             LITRES         0.00

Please enter your command :
 [D] View Daily       [M] View Monthly     [A] View Annual      [C] View Calculate
 [PgUp] Scroll up     [PgDn] Scroll down   [S] Select a Day     [Esc] Exit
```



```
Knight Compressor Stn        REVIEW  CONSUMABLES          Today:87.06.06  17:47

                                                        Hrs since
Unit     Date     Name of Consumable        Units    last change

   1   87.06.05   Lube oil ETO-2380        LITRES        15279.2
   2   87.05.06   Lube oil ETO-2380        LITRES        21217.2
   3   87.05.02   Lube oil ETO-2380        LITRES        30788.7
   4   87.05.06   MPU lube oil             LITRES         5880.0
   5   87.05.05   APU lube oil             LITRES         1128.0

Please enter your command :
 [PgUp] Scroll up    [PgDn] Scroll down                           [Esc] Exit
```

A CRITICAL ASSESSMENT OF THE APPLICATION OF ARTIFICIAL INTELLIGENCE TO ENGINE HEALTH MONITORING

B.D. MacISAAC
GasTOPS Ltd.
OTTAWA, ONTARIO

ABSTRACT

The fundamental elements of artificial intelligence are examined in this paper. A brief description of the current state of the art is provided based on available literature.

The application of artificial intelligence in the form of expert systems is explored and specific prerequisites to the successful application of these techniques are defined. Finally, an examination of how well the various techniques used in the monitoring of gas turbines meets these prerequisites is explored and conclusions are drawn about the applicability of artificial intelligence to this field.

1. INTRODUCTION

The industrial gas turbine, while providing competitively economical power for a specific class of applications, has remained an expensive system to own and to operate. Initial procurement costs for a typical installation are in the order of $300 per KW and annual fuel costs are in the vicinity of $900/KW. It is, thus, no surprise that operators are continuously seeking methods of reducing costs of ownership.

A complete industrial gas turbine involves much more than an engine. The system is "packaged" with all necessary ancillaries including an automatic control and surveillance system to permit automatic operation and full protection of the machinery in all of the common operating environments.

With the development of digital computers continuing unabated, more and more small powerful computers are being embedded in machinery control and surveillance systems. As these processors become more capable (and the systems become more complex and expensive) there is an increasing tendency to request that they perform more and more functions. Thus control and surveillance is being extended to engine health monitoring (EHM) and the trend is clearly towards an extension of these systems to on-line trending, analysis and diagnostics. At this point, there is a real risk of overselling its capabilities. In particular, the current thrust by the U.S. military into artificial intelligence coupled with business interests to capitalize on these developments has led system developers to make claims for which there is doubtful support.

It is within the author's memory that most managers had little if any knowledge of computers. It was a common complaint, therefore, that young computer literate engineers had seduced them into believing that the computer was far more powerful and capable than it actually proved to be. At that time, the computer was not really doing anything particularly clever. Today, the computers aptitude for engineering calculations and payroll cheque writing is taken for granted.

At this time, most managers have some computer literacy and computer designs have more or less stabilized around the so-called "Von Neumann" architecture. One would expect, therefore, that managerial caution would prevail resulting in realistic goals being set for advancement of this technology. This does not seem to be the case. It is well known that computer designs are now quite advanced and are getting cheaper each year. The latest 32 bit PC's have the power of the mainframes of only a few years ago and lack the software to make use of much of this power. It has been by this route that we have entered the information age and now managers are once again being seduced by techno-salesmen who appear to be very knowledgeable in the capabilities of software and software packages.

Hence, the mathematical concepts of artificial intelligence are made to seem just the answer to the problems of dealing with the masses of data available from computers which are currently maintaining surveillance over a wide variety of gas turbines.

It is the purpose of this paper to examine some of the concepts embodied in this technology. This review is presented from the perspective of an analyst knowledgeable in the field of gas turbines and familiar with the real requirement of an operator.

2. ELEMENTARY ARTIFICIAL INTELLIGENCE

In order to assess the applicability of artificial intelligence it is helpful to have some idea of what the words mean in practical terms. In the simplest of terms, artificial intelligence seems to mean the combination of computers and software which results in something which behaves in an intelligent manner. Apart from the fact that a computer is involved (hence the adjective artificial) this is not a wholly satisfactory definition since we have only a poor grasp of the concept of intelligence. It is bound up in "comprehension" and in "quickness of mind"; however, we have at best only a poor grasp of how we think. Thus we look to results to demonstrate rather than explain what we mean. Artificial intelligence therefore has two separate and complementary goals.

a) to understand intelligence
b) to make machines (computers) more useful

Thus, it may be said that the real purpose is to produce a computer model of the human thought process that could be applied to a wide variety of human problems.

To the extent that the development of artificial intelligence can be considered successful, the following basic elements have been identified(1)

a) Heuristic Search Methods
b) Knowledge Representation
c) Logic
d) Common Sense

Heuristic Search Methods

One of the things that we know about humans is that intelligent behaviour often has as much to do with the availability of knowledge as it has to do with the reasoning process. As humans go through life they acquire a vast store of knowledge which is somehow retained in memory. How we recall this information is not at all well understood.

To emulate the process of human recall of information, we have only a computer to work with. Thus, early AI work focused on means of searching for solutions to problems based on constraints usually in the form of "if this is true, then ... else...". To the machinery troubleshooter these are familiar fault trees. As each decision is made the operator is led to another test resulting in another branch point. As he continues down the tree, many different possibilities open up so that the number of branches at the bottom can become enormous. Unless it is a simple problem, a digital computer which must execute everything in sequence will require a substantial amount of time to arrive at an answer.

To reduce the frequency of repeat trys along the wrong path, "rules of thumb" or heuristics were developed to guide the computer along its path. These methods should not be construed as an emulation of a human, they are little more than an expediency until we better understand the mechanisms by which a human recalls information from memory.

Knowledge Representation

The second area where a model of the human mind is required for the development of artificial intelligence is in the matter of representation of knowledge. We know that for a complex problem, a great deal of knowledge must be brought to bear on it. In the case of a human, it might be a piece of knowledge obtained many years previously that holds the clue to the solution. Thus, how to represent this knowledge on a digital computer to most efficiently use it is a real problem and is considered to be a fundamental element of AI.

Logic

The third major element of an artificially intelligent system is logic. How do we deduce something from a set of facts? In other words; how does a human think Again we do not have a very good model for logic processes. Direct implementation of troubleshooting trees is simply a precomputation of a specific set of logic. What to do with more or new information is essentially an unsolved problem although some progress has been made in this regard. The generation of rules from input data is a major area of AI research and is, in essence, the distinguishing feature of expert systems.

Common Sense

Artificial Intelligence researchers found that common sense was an extremely difficult thing to model on a computer. It has been concluded that commons sense is a low level form of reasoning based on a wealth of experience. For example, we have learned to expect that when we drop something, it falls. Similarly, we have learned to anticipate all manner of things in everyday life and we apparently operate in a reflex manner without the need to think much about it. To what extent the human uses common sense in work and how he does it is very poorly understood and represents an area of artificial intelligence which will not likely be solved in the near future.

3. EXPERT SYSTEMS

According to Nilsson (2), there are four recognizable applications of AI as follows.

a) Natural Language Processing

As the name implies, this has to do with developing the ability to talk to a computer and have it interpret your speech and answer you in a natural manner.

b) Computer Vision

In concept, this is similar to speech recognition. It is the development of systems to let a computer see, to identify objects and to "understand" the significance of 3-D space, of shading and of colour.

c) Expert Systems

This is the development of the computer to behave like an expert on a specific topic, to interpret specific circumstances in a diagnostic sense.

d) Problem Solving and Planning

This might be regarded as the use of an expert system concept to develop a solution approach to a specific problem.

Natural language processing and computer vision are the life blood of the robot age; however, it is a concept of the expert system which intrigues most of us. It is consequently regarded as the hottest topic in AI today because if it can be made to work, it has an enormous range of applications.

The major elements of an expert system are shown in Figure 1 (3).

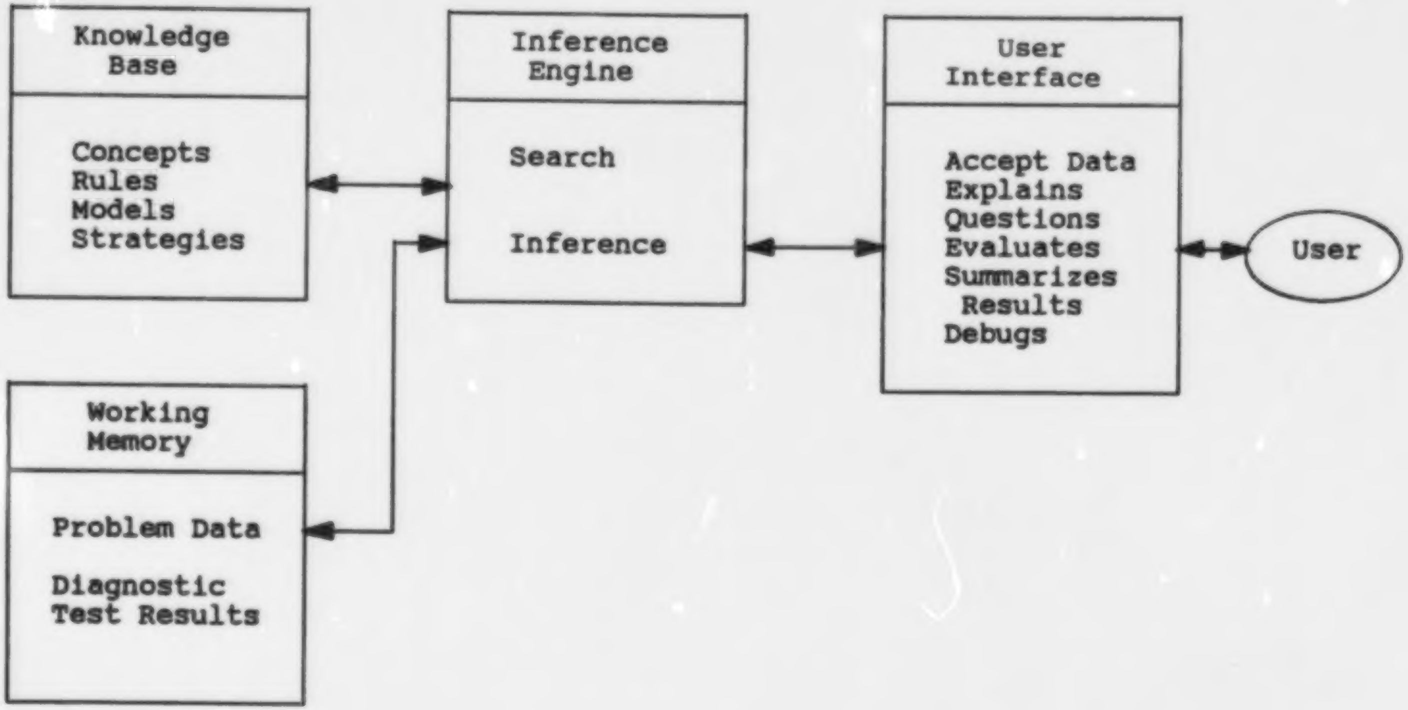

Knowledge Base
Concepts
Rules
Models
Strategies
Working Memory
Problem Data
Diagnostic Test Results
Inference Engine
Search
Inference
User Interface
Accept Data
Explains
Questions
Evaluates
Summarizes Results
Debugs
User

Figure 1 Components of an Expert System

The user interface provides a number of functions or capabilities as noted. These include the collection of data, experimenting with the knowledge base, running test cases, interrogating the user for more data and providing output information. In the context of EHM, this presumably means that on-line data collection is simply one form of input with other inputs coming from an operator on the scene.

The heart of the expert system is the inference engine. This component of software guides the search and infers from the current data and the results of the search that an answer has been obtained or best course of action found. Thus, it is this component which evaluates the rules and generates new rules from new data. In short, it is this element that attempts to emulate the human thinking and learning process.

The knowledge base is that software which contains the information pieced together about the system upon which the inference engine can work together with the current data to obtain an answer or diagnosis. The knowledge base is usually considered to consist of levels of information as follows.

Concepts are little more than declarations of the various physical entities involved in the subject. For example, it would deal with the components of the gas turbine as compressors, turbines, etc. These would then be broken down further to indicate individual elements like shafts, disks blades, etc. and would further define how they are connected together and how the engine works.

Rules are the collected diagnostic wisdom of the human experts. They are either calculated from cause/effect relationships or are known empirically from experience. They are, in short, the troubleshooting trees or fault matrices normally associated with machinery diagnosis. For example, if one could say

IF fuel consumption is high,
THEN the probable cause is ....

This would form a usable diagnostic rule.

Models are collections of interrelated rules. For example, they might be a chain of troubleshooting rules associated with a specific component such as a fuel control.

Strategies are procedures used to assist in the use of the knowledge base. They assist in guiding the search and in some cases attempt to resolve conflicts where several plausible diagnoses are possible. This latter situation may take the form of asking the human operator to check something to obtain additional data.

A partial rule network for fault isolation of a gas turbine is shown in Figure 2.

Prerequisites to Expert Systems

To construct an expert system for any application it is generally agreed (ref. 1, 4) that the following are prerequisites to success.

a) There must be at least one human expert acknowledged to perform the task well.

b) The primary source of the expert's exceptional performance must be special knowledge, judgement and experience.

c) The expert must be able to explain the special knowledge and experience and the methods he uses to apply them to the particular problem.

d) The task must have a well bounded domain of application. Duda (4) goes on to say:

> "Many scientific and technical tasks meet these conditions, especially where there is a tradition of consultation. Certain kinds of tasks are not good subjects for knowledge-based expert systems. For example, although a mathematician possesses specialized knowledge, the additional knowledge needed to function in that role is far more extensive than current knowledge-based expert systems can handle. On the other hand, where there are well-defined mathematical procedures for solving a problem, knowledge-based expert systems are unnecessary. For most perceptual problems, experts are unusual, relevant knowledge is extensive but difficult to pinpoint, and general understanding of the perceptual process is limited. And many tasks are still waiting for the first expert to appear. An example is earthquake prediction. There are now no geologists who can predict earthquakes accurately. An expert system for earthquake prediction should not be expected to fare any better than the human sources of its knowledge base."

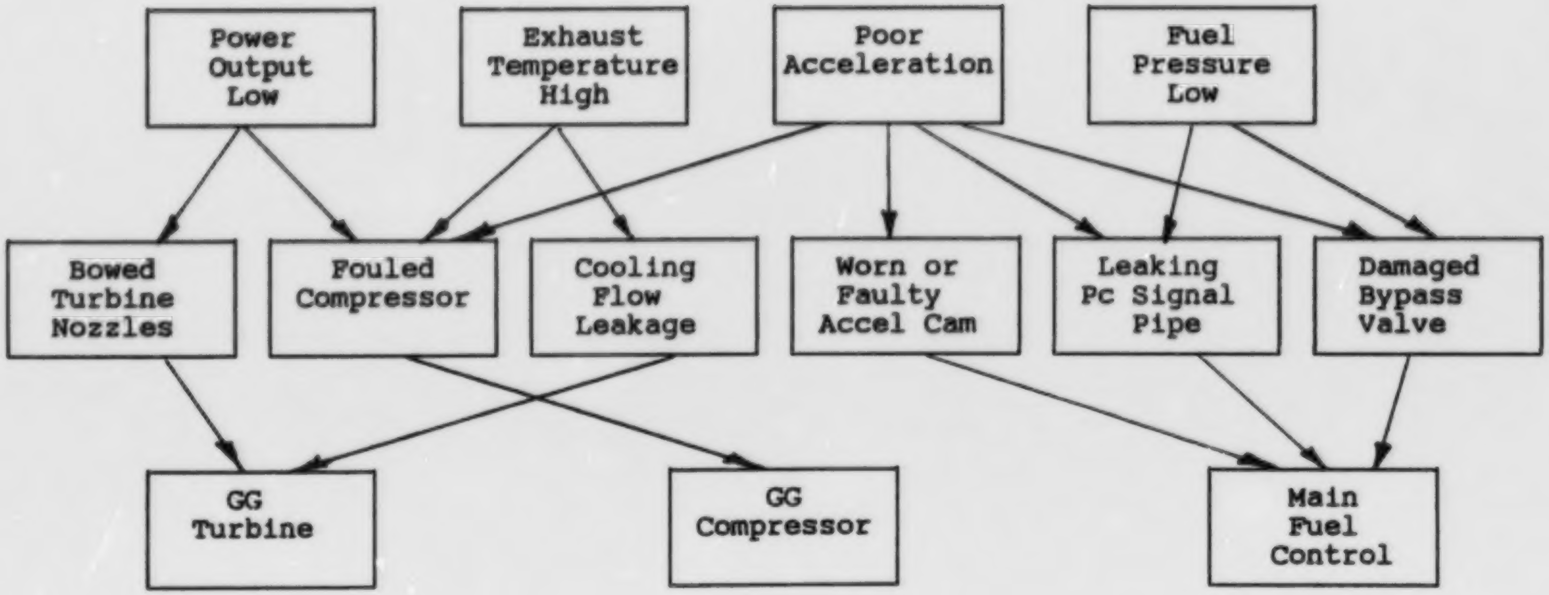

Power Output Low
Exhaust Temperature High
Poor Acceleration
Fuel Pressure Low
Bowed Turbine Nozzles
Fouled Compressor
Cooling Flow Leakage
Worn or Faulty Accel Cam
Leaking Pc Signal Pipe
Damaged Bypass Valve
GG Turbine
GG Compressor
Main Fuel Control

Figure 2   Partial Rule Network for Troubleshooting

Based on the above prerequisites to building a successful expert system, the question remains whether or not gas turbine health monitoring analysis and diagnostics have reached the state of development where artificial intelligence can be used successfully to replace a human expert.

4. THE ELEMENTS OF ENGINE HEALTH MONITORING

The decision to apply engine health monitoring techniques to a fleet of gas turbines is largely a managerial problem. The incentive for doing so can be purely economic (phrased in $/KW) in which case engine performance holds some importance or it may be safety of people or equipment in which case the mechanical health of the engine is paramount. It is beyond the scope of this paper to discuss the various aspects of EHM from this perspective; however, it is worth noting that a well managed EHM system will impact on all aspects of support including spare parts inventories, scheduling of repairs, and a host of engineering activities ranging from design of repair schemes to analysis supporting future equipment purchases.

While these are of great importance to the economic effectiveness of EHM in general, they are beyond the domain of an expert. In other words, this global view of EHM results in a poorly bounded situation for which an expert system would fair poorly. We must therefore focus on the technical side of the EHM question; the ability to comment accurately on the physical condition of the engine. To do so it is necessary to examine the various techniques that are used in EHM.

A review of EHM techniques (5) results in a somewhat daunting array of specialized topics. Table 1 presents the more common ones in use today. Each will be discussed briefly in the following sections.

Event Recorders are typically embedded in automatic control and surveillance systems or they may involve a human operator. Their purpose is to provide maintenance personnel with a starting point in a troubleshooting exercise. Well designed event recorders provide, through the normal mix of on-line instrumentation, a record of conditions leading up to the event, the event itself and for some time past the event. This means that when one event triggers a series of events (e.g. limit exceedances) the order of occurrence is captured which has often proved useful to the maintenance personnel.

Table I

Primary EHM Techniques

EHM Technique	Area of Concern	Primary Purpose
Event Recorders	All	Troubleshooting On-site Repairs
Vibration Monitoring	Bearings	Damage Assessment
Boroscope	Gas Path	Damage Assessment
SOAP	Lube System	Wear
Ferrography	Lube System	Wear
Debris Monitoring	Lube System	Wear
Time/Temperature	Hot End	Parts Life
LCF Counters	High Stress Components	Parts Life
Gas Path Analysis	Gas Path	Performance Assessment

In general, troubleshooting involves known cause/effect relationships and manufacturers frequently publish troubleshooting manuals to assist in isolating the trouble. Clearly, it requires a very large body of experience to have been built up over a long period of time. Thus one can expect to find bonafide experts in such an application. The work is frequently restricted to those components which are repairable in the field; thus it is a well bounded problem.

In concept, troubleshooting using known cause/effect relationships and event recorders is an area of EHM to which expert systems can be applied. In fact, there is quite a number of such expert systems under development (see for example ref. 6, 7, 8, 9, 10). It is not yet clear how successful they will be nor is it obvious that it is the correct approach. Nevertheless many commercial organizations are interested in developing this capability.

Vibration Monitoring is perhaps the oldest and best established on-line EHM technique. Originally, it was intended simply as a means of early recognition of a failed bearing with the express intention of containing the damage. As the gas turbine has become more sophisticated with flexible bearing mounts and squeeze film bearing dampers, it has become increasing difficult to obtain a clear cut indication of bearing distress. The technology has thus been extended to very sophisticated analysis of vibration signatures. Stewart (11), a recognized expert in this area suggests caution in applying AI techniques to this area. He comments as follows:

> "The primary lesson learned has been what enormous difficulties there are in developing reliable techniques for the more sophisticated types of machine. This was particularly so for vibration analysis. For example, it took the company 8 years to approach simple consistency (let alone guaranteed performance) on the matter of gear tooth damage detection."

This suggests that we have some ways to go to establish the bounds of this problem. If it involves some form of pattern matching it is quite likely well beyond the capability of current computers and expert systems technology. While one should not rule out the future use of AI in this area we are clearly some distance away from a solution.

Boroscoping is a well known method of inspection which allows a trained man to view specific internal components for damage. It is a manual technique although some systems permit photography and closed circuit TV.

The application of AI to this area will involve matching of recorded images with images of known damage patterns. It is, in short, a form of computer vision. This technology is regarded by AI researchers as being in its infancy. The ability to recognize common household shapes like a cup and saucer is the current limits of this technique. The ability to discriminate between boroscope images is clearly a long way off.

SOAP or Spectrometric Oil Analysis Program involves the periodic sampling of oils to assess their contamination levels from minute wear particles. While it is an established technology to determine the contamination levels, systematic and consistent diagnostic techniques in SOAP remain an elusive goal. Experts do not always agree about the results of the analysis and frequently have difficulty isolating the source of increased wear.

AI may be helpful in collecting and analyzing results from a research viewpoint; perhaps the establishment of recognizable patterns, however, the use of AI in an on-line EHM sense cannot realistically be developed until there is consensus among experts and a better knowledge base is established for the application.

Ferrography is a second oil analysis technique which specifically looks for ferromagnetic particles in lubricating oil. All of the arguments which apply to SOAP apply in equal measure to Ferrography. It is much too early to have established reliable troubleshooting techniques using this technique.

Debris Monitoring is the simplest form of oil analysis and is regarded by many as the most reliable. It has several forms; magnetic plug chip detectors, filter analysis and more recently on-line, full flow debris monitors. The first two involve removal and inspection. In this regard a human operator sees quantity and pattern. The third is a more recent development and provides a quantitative reading of particle density in the flow. It is thus a new sensor.

The use of expert systems seems unlikely for magnetic plug and/or filter analysis since it requires a human to quantify what he sees. Furthermore, it would inevitably involve camera imagery and computer vision. For on-line debris monitoring, the use of expert systems may well be possible. Certainly the reading is simple and direct. The existence of correlations of cause/effect have been at least partly established; however, more research work must be done to fully develop the knowledge base.

LCF Counters and Time/Temperature Records are closely related. The former involves a means of counting stress cycles due to speed changes within the power envelope while time/temperature recorders count thermal cycles. The express purpose is to apply these data to an algorithm in order to estimate the rate of usage of engine parts life. Newer airborne engines use these data to track engine usage and to schedule removal of lifed parts.

Both LCF and Time/Temperature counters are very global representation of extremely complex metallurgical phenomena. Research continues to improve our ability to predict crack propagation and hence life of parts. It is a controversial subject and our level of knowledge of it is probably similar to our level of knowledge of earthquakes. Experts in this complex field are those who know most about out; however, they do not meet the criteria of an expert as defined by Duda (4). At the moment, our best methods of life prediction are not better than a factor of 2 on cold end parts and a factor of 5 on hot end parts (12).

Application of Expert Systems to the field of LCF monitoring is completely inappropriate at this stage. Based on our current level of knowledge, a good bookkeeping system is the primary requirement; relational database systems are currently is use. Continuous research is required in this field in order to better understand the phenomena involved. This will directly aid the designers and indirectly aid operators.

Gas Path Analysis is the last major area of EHM. It involves the analysis of measurements taken in the gas path in order to estimate current performance. These estimates are then compared to baselines to assess deviations. Interpretation of the deviations in a diagnostic sense has been aided by the use of fault modelling techniques; however, far too little is known about the individual cause/effect relationships to establish a diagnosis with any certainty. Certainly models indicate that it is possible to isolate faults to a specific component of the engine; however, many of the fault models await experimental evidence of their validity before the next step can be taken. At present, it is possible to precompute many cause/effect relationships and to store these "fault signatures" in a computer for on-line comparison. There is, however, much ambiguity in the process and probabilities have to be assigned to the diagnosis.

In general, performance analysis is poorly understood by many operators, however, there is no disagreement among the experts as to the methods to be employed. Thus, this field is a candidate for future expert system applications. The major stumbling block to its current use is the lack of a knowledge base. In other words, fault signatures must be established by a combination of analysis and experiments. Since every engine is different, this must be undertaken at a fundamental (stage) level and from there generalized methods can be evolved. Figure 3 represents a generalized approach to engine performance monitoring. The reader should note that a major element of this is a continuous development of the knowledge base in order to achieve consistent and reliable diagnosis.

5. DISCUSSION

Examination of the field of EHM reveals that there is no one expert. It is, in fact, a field comprising a number of technologically complex subjects which, in many respects, bear little resemblance to each other. Techniques are continuing to evolve suggesting that it is not a static technology. Continuous research is clearly required in several areas in order to establish the validity of the techniques.

The application of artificial intelligence to the field of EHM raises more questions than it provides answers. For example, does it require a single unified expert system or a number of tailored systems each with its own peculiarities? How many of the current EHM techniques are suited for adaptation to an expert system? Clearly LCF counters do not and those techniques which rely upon human visual perception cannot be included at this time. This includes boroscoping and several oil analysis techniques. Similarly , which techniques have reached the level of maturity that suggest a number of experts can consistently produce a correct diagnosis? In other words, does a real expert within the narrow definition of AI actually exist? Vibration monitoring, and gas path analysis are still in a state of developing the knowledge base. They may eventually be candidates but not now. All of this seems to suggest that the event recorder/troubleshooting trees is the only likely candidate for an expert system. None of the others seem ready at this time.

The foregoing discussion focussed on the major EHM techniques and their applicability to an expert system approach. The readiness of the expert system software to

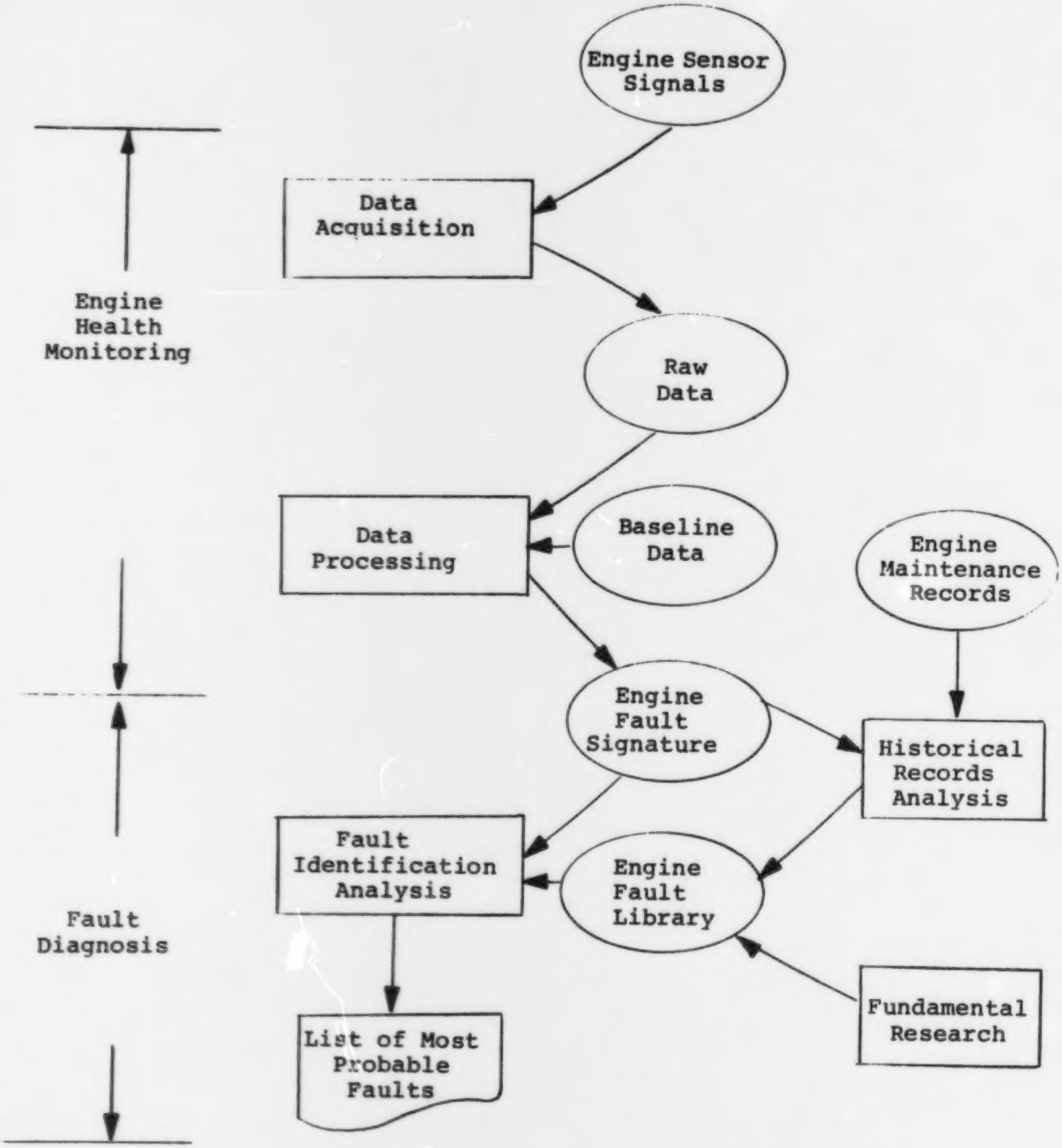

Engine Sensor Signals
Data Acquisition
Engine Health Monitoring
Raw Data
Data Processing
Baseline Data
Engine Maintenance Records
Engine Fault Signature
Historical Records Analysis
Fault Identification Analysis
Engine Fault Library
Fault Diagnosis
Fundamental Research
List of Most Probable Faults

Figure 3 Engine Health Monitoring

handle the problem is a major concern. Despite the thrust of the development of practical expert systems provided by the U.S. military, there is no consensus in the literature that such systems are highly developed. A number of comments are noteworthy.

> "By the time an AI language program is adapted to run a real problem, it is so tuned to that particular problem that most of the flexibility is gone."
>
> (Matyskiela, TRW - Ref. 6)

> "Performance data on operational AI systems are nearly nonexistent. In fact, many professionals who purport to develop "high performance" systems seem genuinely puzzled by the query, "Can I see the performance data?" Often there is no data. The "success" of a system is in the mind of the builder."
>
> (Harris & Owens - Ref. 13)

A final comment on the subject of EHM and the need for experts in an age of technology is important. One cannot help but notice that an area of technology that computers have had enormous and successful impact on is communications. We have developed very sophisticated means of transmitting large quantities of data over thousands of miles. Strides in this area are being made every year. We are on the verge of retiring well established microwave technology which was thought to be very fast little more than a decade ago. Indications are that we will be replacing it with particle beam (light) carriers which will improve band widths enormously. In short, we will be able to transmit data at speeds approaching that of light. Furthermore, these are proven technologies at a stage of development where they can be commercialized.

So why not a human expert? Surely, with the communications systems available today, a team of human experts could be made available to a wide variety of EHM situations. They would continue to learn and to develop the expertise. They could provide a learning ground for other experts. It may not be so futuristic as artificial intelligence, but it might be more cost effective.

6. CONCLUSIONS

An examination of prerequisites to building an expert system has been made. Based on these prerequisites and the current state of the art of EHM the following conclusions can be drawn.

a) Only conventional event records/troubleshooting trees meet all of the prerequisites for AI.

b) Vibration Monitoring and Gas Path Analysis require substantial development of the knowledge base before AI methods can be applied. Even then, the magnitude of the problem may be more than current computers can handle.

c) Boroscoping and several oil monitoring techniques require human visual perception to be effective. These are currently beyond the capability of AI systems.

d) Mechanical and thermal fatigue counters are a bookkeeping problem and are currently inappropriate for AI methods.

e) Modern communications technology together with human experts may be a more cost effective approach.

References

1. Gevarter, William B.
"Artificial Intelligence, Expert Systems, Computer Vision and Natural Language Processing"
Noyes Publication, New Jersey

2. Nilsson, N.J.
"Artificial Intelligence: Engineering, Science or Slogan"
A.I. Magazine, Vol. 3, No. 1
Winter 81/82

3. Rychener, Michael D.
"Expert Systems for Engineering Design"
Expert Systems, Vol. 2, No. 1
January 1985

4. Duda, Richard O. and Gaschnig, John G.
"Knowledge-Based Expert Systems Come of Age"
BYTE Publications
September 1981

5. MacIsaac, B.D.
"A Review of Engine Health Monitoring Techniques"
4th NRCC Symposium on Gas Turbine Operations and Maintenance
Toronto, 1981

6.
"Technical Survey: Artificial Intelligence:
Aviation Week & Space Technology
February 17, 1986

7. Johnson, Harold E., Bonissone, Piero P.
"Expert System for Diesel Electric Locomotive Repair"
J. of FORTH Application & Research, Vol. 1, No. 1
January 1984

8. Smiley, R. Gene
"Expert Systems for Machinery Fault Diagnosis"
Sound & Vibration
September 1984

9. Nelson, William R.
"Reactor: An Expert System for Diagnosis and Treatment of Nuclear Reactor Accidents"
Proc. of the AAAI, 1982

10. Lance, Nick and Malin, Jane T.
"An Expert Systems Approach to Automated Fault Diagnosis"
SAE Paper 851380, 1985

11. Stewart, R.M.
"Machinery Management Based on the Application of Modern Diagnostics", Control Theory and Artificial Intelligence",
Proc. of International Conf. on Conditioning Monitoring
April 1984

12. Kirkhope, J. and Muir, D.E.
"Low Temperature Effects on Aero Gas Turbine Engine Durability"
GasTOPS Ltd. Report GTL-7-8-TR.2
DND Contract 2SV82-00068
March 1983

13. Harris, S.D. and Owens, J.M.
"Some Critical Factors that Limit the Effectiveness of Machine Intelligence Technology in Military Systems Applications"
Naval Aerospace Medical Research Lab.
Special Report 84-2, 1984

Predicting the Effects of Progressive Compressor Fouling on Gas Turbine Performance

by
G.F. Aker
Research Assistant
and
H.I.H. Saravanamuttoo
Professor
Department of Mechanical and Aeronautical Engineering
Carleton University
Ottawa, Ontario

Abstract

A computer simulation program was developed employing the stage stacking technique in an effort to quantify the effects of compressor fouling on gas turbine performance. By matching the compressor maps generated by the simulation software to generalized turbine characteristics, the effect of progressive compressor fouling on several operating parameters was predicted. The results of these simulations can be used to correlate changes observed in monitored parameters, to changes in heat rate and power output, thus allowing the user to determine the most economical interval between compressor washes. The engine used in the simulation was the 2860 kW Solar Centaur T-4000.

Introduction

The fouling of gas turbine compressors remains one of the most common causes of lost power and thermal efficiency facing users today [1]. Many users have recognized this and adopted the practice of performing regularly scheduled compressor washes [2]. This can go a long way towards limiting the effect of fouling on operating costs but given that fouling is very much a factor of site conditions, engine condition and engine type, it would not appear to be the optimum solution [3]. Performing compressor washes too frequently not only increases the maintenance costs of the engine but can, in time, erode compressor blade surfaces.

In an attempt to provide engine users with a means of determining the optimum interval between compressor washes, Carleton University has developed gas turbine simulation software to simulate the effect compressor fouling has on engine operating parameters and performance. It is hoped that this type of work will not only provide information on how engine performance is affected by fouling, but also enable the development of an "on condition" compressor cleaning capability based on trend analysis of monitored parameters.

Method

Simulating Compressor Performance

The simulation software used in this study employes the stage stacking technique to simulate compressor performance and generate compressor maps. The theory behind the stage stacking technique has been well documented and so only a brief overview will be presented here. Further detailed information is available in references [4][5][6].

The stage stacking technique used in this study involves estimating the performance of individual compressor stages on the basis of a generalized stage characteristic. The generalized characteristic consists of pressure and temperature coefficients presented as a function of a flow coefficient. Once the flow through a stage has been estimated making use of the annulus dimensions and stator exit angles, the flow coefficient for the stage can be evaluated. This flow coefficient and the local flow properties then define the stage pressure ratio and temperature rise. The stage pressure ratio is then multiplied by the stage inlet pressure and the temperature rise is added to the stage inlet temperature to provide the stage exit conditions. These stage exit conditions then represent the inlet conditions to the next stage where the process is repeated until all of the compressor stages have been "stacked". The stage exit conditions from the last stage represent the compressor delivery parameters. The compressor is stacked at various mass flow rates between the surge and choke flow rates for the compressor at a specified speed. This defines a constant speed line for the compressor being simulated. The compressor is restacked until the 100% non-dimensional speed line intersects the design mass flow rate and pressure ratio specified by the manufacturer. The predicted compressor map for the Solar Centaur T-4000 gas turbine is presented in Figure 1.

The stage stacking technique provides a systematic and logical method for evaluating the effects of compressor fouling on compressor performance by allowing the performance of each individual stage to be modified to reflect the fouled condition. The effect of this stage performance deterioration on the performance of the compressor and engine can then be evaluated.

The performance of the entire engine is simulated by matching the compressor maps generated using the stage stacking technique to generalized turbine characteristics [7]. Clean engine performance is simulated by modifying the maximum flow parameter on both the gas generator and power turbine characteristics until the design specifications of the engine are met. A comparison of the design point specifications and the clean simulation results for the Centaur is presented in Table 1.

Simulating Compressor Fouling

The fouling of the compressor stages was simulated by reducing the flow and efficiency of the stage characteristics. Previous investigations into simulating compressor fouling have validated this approach [8][9][10]. In order to simulate the progressive nature of compressor fouling and its resultant effect on compressor and engine performance, a linear fouling model was developed. This model simulates compressor fouling by assuming the fouling process occurs in steps, where each "step" represents a 1 percent reduction in the flow characteristic of the affected stages. Each step also increases the number of affected stages by 1. Using this method, the onset of fouling (level 1) is simulated by a 1 percent reduction in the flow characteristic on the first stage only. Similarly, moderate fouling, (level 3) is simulated by a 3 percent reduction in the flow characteristic on the first stage, a 2 percent reduction on the second stage and a single percent reduction on the third stage with the remaining compressor stages being left unmodified.

In order to present the results of this study in a comprehensive manner, the stage efficiency characteristic was modified as a function of the flow characteristic modification. This is denoted by the designation (1:X) where "1" represents the one percent reduction in the flow characteristic per fouling level and the "X" represents the corresponding reduction in the efficiency characteristic in percent per fouling level. In other words, the designation

(1:0.75) means that every progressive level of fouling is simulated by a one percent drop in the flow characteristic and a 0.75 percent drop in the efficiency characteristic.

It is believed that the linear fouling model provides a reasonable representation of the progressive nature of fouling up to the point where half of the stages in the compressor are fouled. Beyond this point, it is likely that there is a greater build up of contaminants at the front of the compressor which would not be accurately represented by this model.

Results

Before simulating the fouling phenomenon, it was necessary to validate the clean engine performance curves generated by the simulation software. This was done by comparing the results from a "zero time" engine test performed on a Centaur at Nova's Meikle River compressor station, with performance curves generated using the simulation software. As can be seen from the results of this comparison presented in Figures 2 to 6, there was excellent agreement between the test and the simulation results.

Having demonstrated the validity of the computer model, various simulations were run to determine the effect of progressive levels of compressor fouling on power output and engine heat rate. The effects on typically monitored parameters such as compressor delivery pressure (CDP), inter turbine temperature (ITT) and mass flow rate were also recorded. The results of these simulations are presented in Figures 7 to 11.

Discussion

The effect of the fouling simulation on the Centaur's performance is presented in Figures 7 and 8. The results are presented as the change from the clean simulation values in percent, at constant non dimensional gas generator speed. The power output was by far the most significantly affected by the simulated fouling. It can be observed that as compressor efficiency decreases there is less fall off in power with fouling. This is a direct result of the increased ITT due to loss of compressor efficiency. The effect of the compressor efficiency on ITT is shown in Figure 9. It has been assumed for the purposes of this study that any constraint on the turbine inlet temperature has not been exceeded as a result of the simulated fouling, however, if the TIT limit had been exceeded necessitating a reduction in gas generator speed, the reduction in power would be even greater.

The simulated fouling also affected the engine's heat rate as illustrated in Figure 8. While the heat rate was not severely affected by the reduction in compressor stage efficiency, it is important to note that the heat rate increased as the stage efficiency was reduced. Of far greater importance however, is the fact that the heat rate increased by 1.5 percent over its clean value, at the third level of simulated fouling.

It is apparent from these results that compressor fouling can significantly affect both power output and heat rate at quite low fouling levels. It therefore becomes important to determine if these low levels of fouling can be detected by variations in typically monitored parameters. The changes in mass flow rate and compressor delivery pressure were also recorded during the fouling simulations. The changes noted in these parameters are presented in Figures 10 and 11 respectively.

From these graphs it can be seen that the CDP has decreased by approximately 1.6 percent at the third level of fouling, while the mass flow rate has dropped by approximately 1.5 percent. The ITT shows little in the way of consistent change, and may increase or decrease depending on whether there is a change in compressor efficiency. If there were no drop in compressor efficiency, the compressor would rematch at a lower pressure ratio and turbine inlet temperature due to the reduction in flow. It would therefore appear that ITT is of limited value as an indicator of compressor fouling.

Accurately measuring and monitoring changes in the compressor delivery pressure of 1.5 percent is somewhat optimistic at present even using expensive computer controlled monitoring systems. However, Scott [2] reported the successful use of intake depression monitoring to detect fouling in a study done at TransCanada PipeLines. The system described by Scott had the ability to detect changes in mass flow rate in the order of 1 percent. In light of the fouling simulation results presented in this paper, it would appear that Scott's method of inferring deterioration in mass flow rate could be very beneficial in detecting compressor fouling before drastic increases in heat rate occur.

Conclusions

The stage stacking technique provides a valuable tool for the systematic investigation of the effects of compressor fouling on both compressor and engine performance. A linear fouling model was introduced to simulate the progressive nature of compressor fouling in gas turbine compressors by modifying the stage flow and efficiency characteristics. It is believed that this model is a reasonably accurate representation of the fouling phenomenon up to the point where approximately fifty percent of the compressor stages have been affected. It was also determined that an engine's power output and heat rate are significantly affected at very low levels of compressor fouling. While the CDP and mass flow rate can be considered to be reliable indicators of compressor fouling, the change in these parameters may be too small to be used effectively, and changes in inlet depression may be the most effective approach.

Acknowledgements

The authors would like to thank GasTOPS Limited, Ottawa for their assistance in setting up the compressor stage stacking model. The financial support of NSERC is gratefully acknowledged.

References

[1] Upton,A.W.J., "Axial Flow Compressors and Turbine Blade Fouling: Some Causes, Effects and Cleaning Methods", First Symposium on Gas Turbine Operations and Maintenance, National Research Council of Canada, Edmonton, Alberta, Oct. 1974.

[2] Scott, J.N., "Axial Compressor Monitoring by Measuring Air Intake Depression", Third Symposium on Gas Turbine Operations and Maintenance, National Research Council of Canada, Halifax, Nova Scotia, Sept 1979.

[3] Williams, L.J.,"The Optimisation of Time Between Overhauls for Gas Turbine Compressor Units", Fourth Symposium on Gas Turbine Operations and Maintenance, National Research Council of Canada, Toronto, Canada, Sept. 1981.

[4] Stone, A. "Effects of Stage Characteristics and Matching on Axial Flow Compressor Performance.", Trans. ASME Aug. 1958, pp 1273-1293.

[5] Howell, A.R. and Calvert, W.J.,"A New Stage Stacking Technique for Axial-Flow Compressor Performance Prediction",Trans. ASME v100 Oct.1978, pp 698-703.

[6] Southwick, Robert D.,"A Stage Stacking Technique for Axial-Flow Compressors with Variable Geometry,"ASD WPAFB, ASD-TR-74-38, Oct. 1974.

[7] Cohen,H., Rogers, G.F.C. and Saravanamuttoo, H.I.H., *Gas Turbine Theory*, Third ed. Longmans 1987.

[8] Saravanamuttoo, H.I.H., and MacIsaac, B.D.,"Thermodynamic Models for Pipeline Gas Turbine Diagnostics", Trans. ASME, J of Eng for Power, v 105, n4, Oct 1983, pp 875-884.

[9] Saravanamuttoo, H.I.H. and Lakshminarasimha, A.N. "A Preliminary Assessment of Compressor Fouling", ASME 85-GT-153, March 1985

[10] Lakshminarasimha, A.N. and Saravanamuttoo, H.I.H., "Prediction of Fouled Compressor Performance Using Stage Stacking Techniques", ASME Symposium on Turbomachinery Performance Deterioration, FED-Vol 37, May 1986.

Table 1. Solar Centaur Design Point Specification Match

Parameter	*Units*	*Literature*	*Simulation*	*Percent Difference*
Mass Flow	$\frac{Kg}{s}$	17.23	17.23	0.00
CDP	KPa	911.7	911.7	0.00
Power	KW	2856	2857	−0.04
Heat Rate	$\frac{MJ}{KWHR}$	13.58	13.30	−2.06
EGT	°C	427	428	1 °C

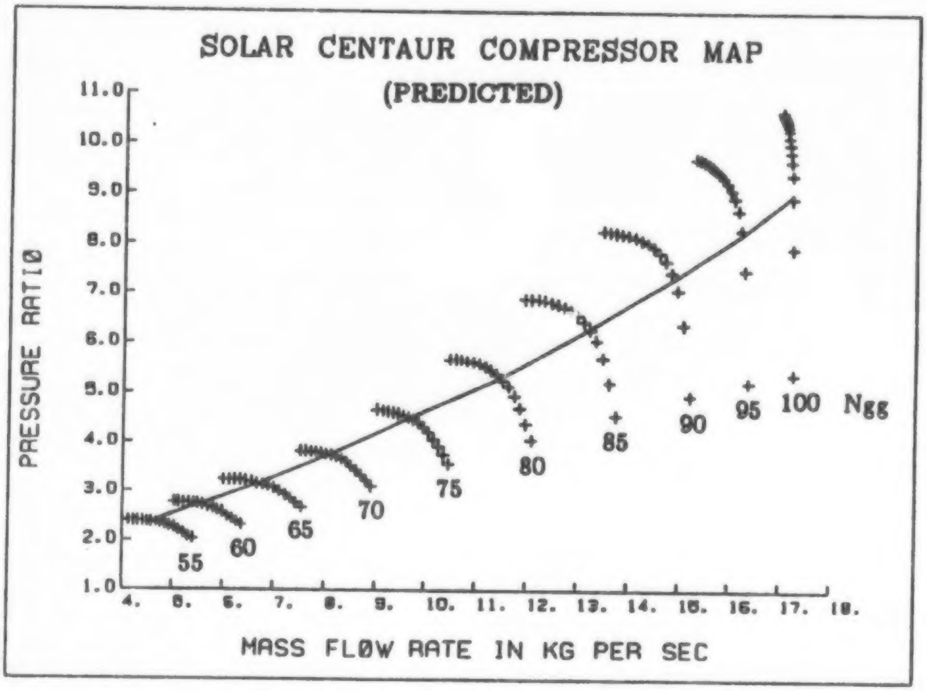

SOLAR CENTAUR COMPRESSOR MAP
(PREDICTED)
PRESSURE RATIO
11.0
10.0
9.0
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.0
4.
5.
6.
7.
8.
9.
10.
11.
12.
13.
14.
15.
16.
17.
18.
MASS FLOW RATE IN KG PER SEC
55
60
65
70
75
80
85
90
95
100
Ngg

Figure 1

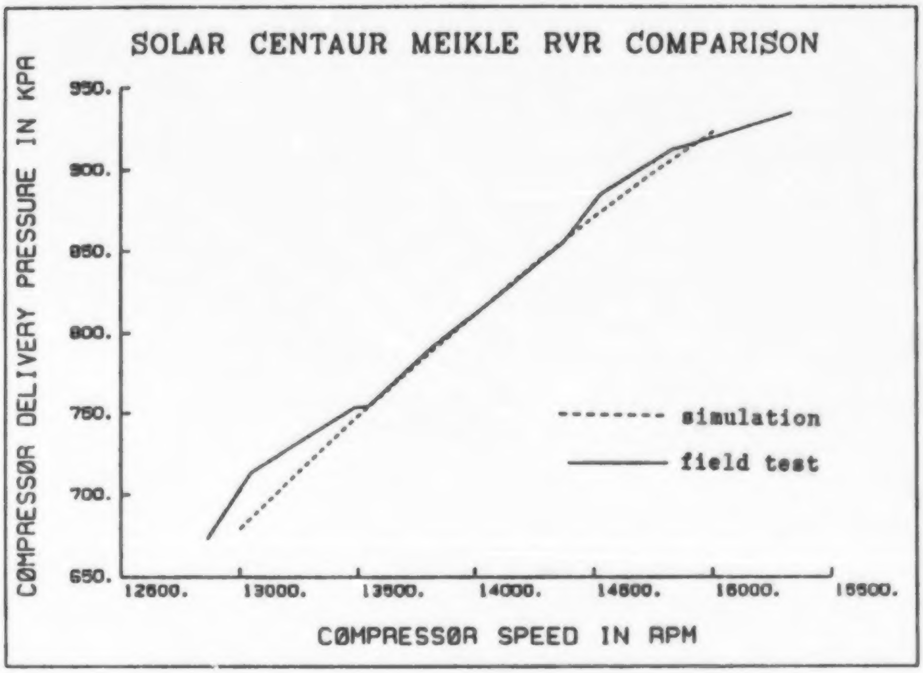


Figure 2

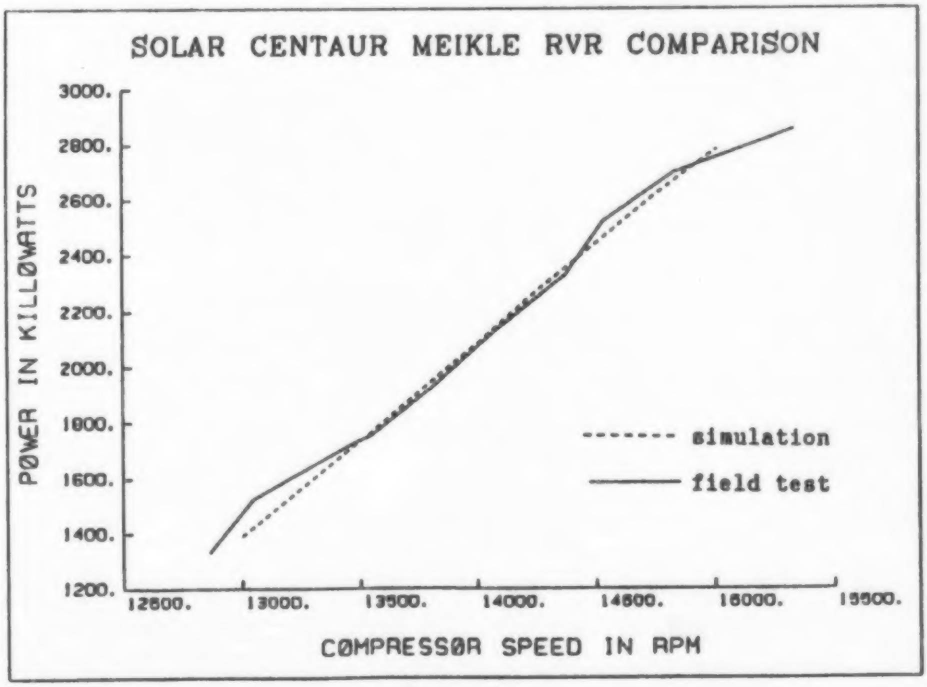


Figure 3

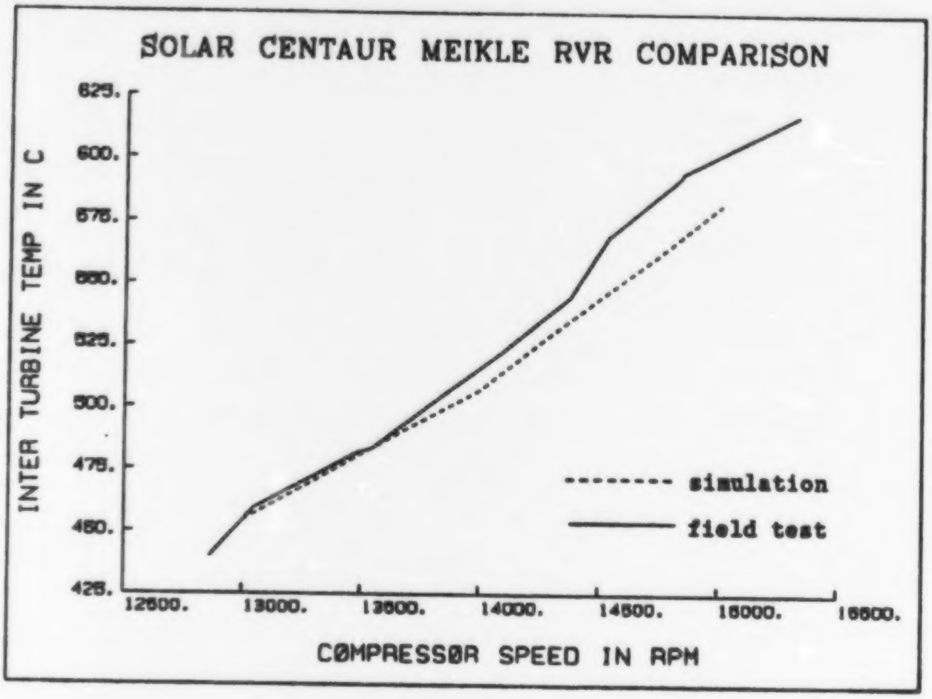


Figure 4

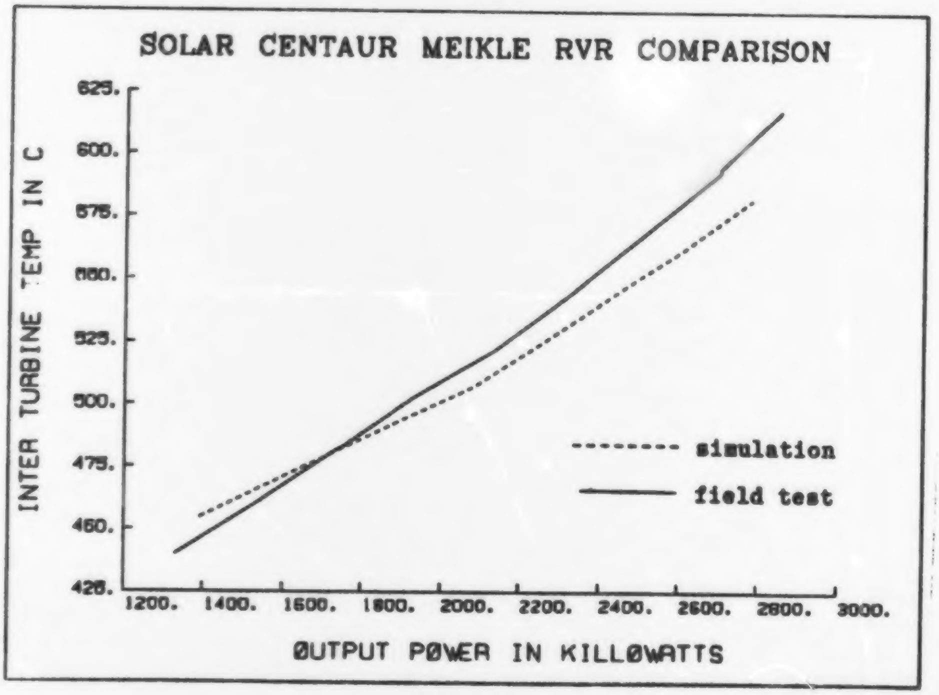


Figure 5

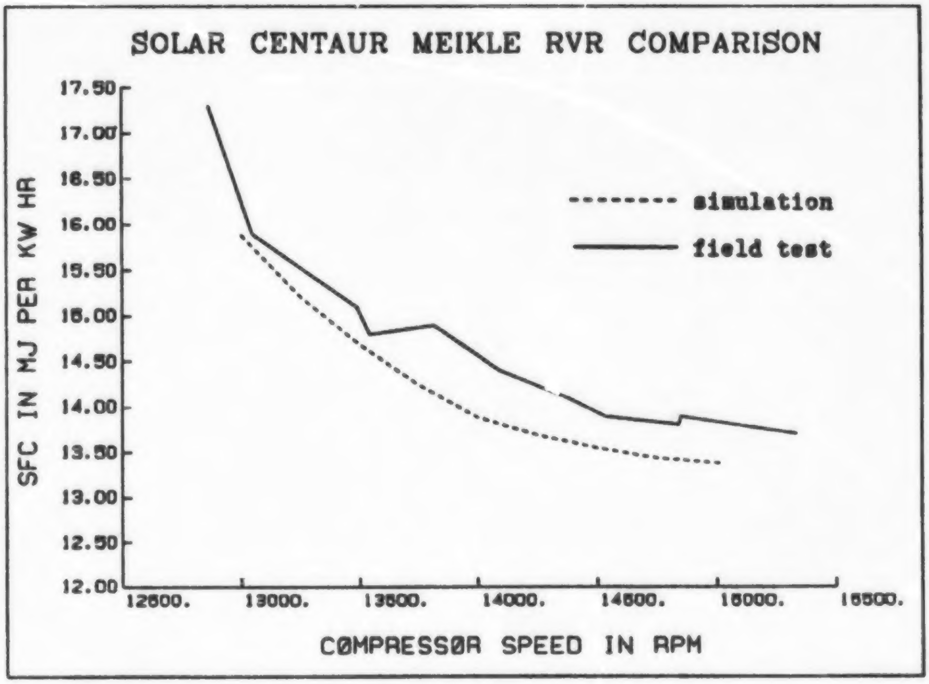

SOLAR CENTAUR MEIKLE RVR COMPARISON
SFC IN MJ PER KW HR
17.50
17.00
16.50
16.00
15.50
15.00
14.50
14.00
13.50
13.00
12.50
12.00
simulation
field test
12500.
13000.
13500.
14000.
14500.
15000.
15500.
COMPRESSOR SPEED IN RPM

Figure 6

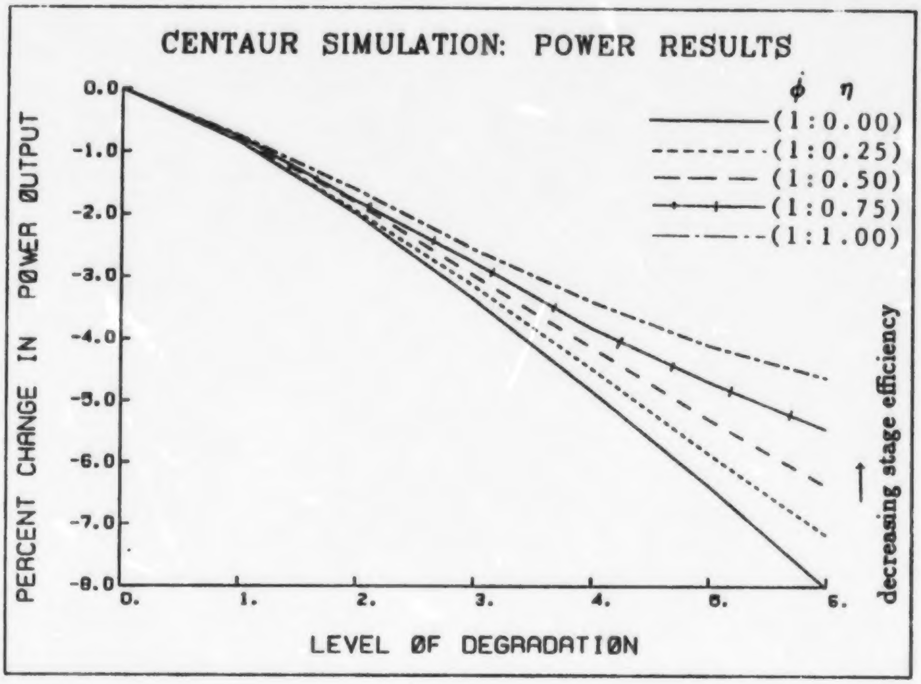


Figure 7

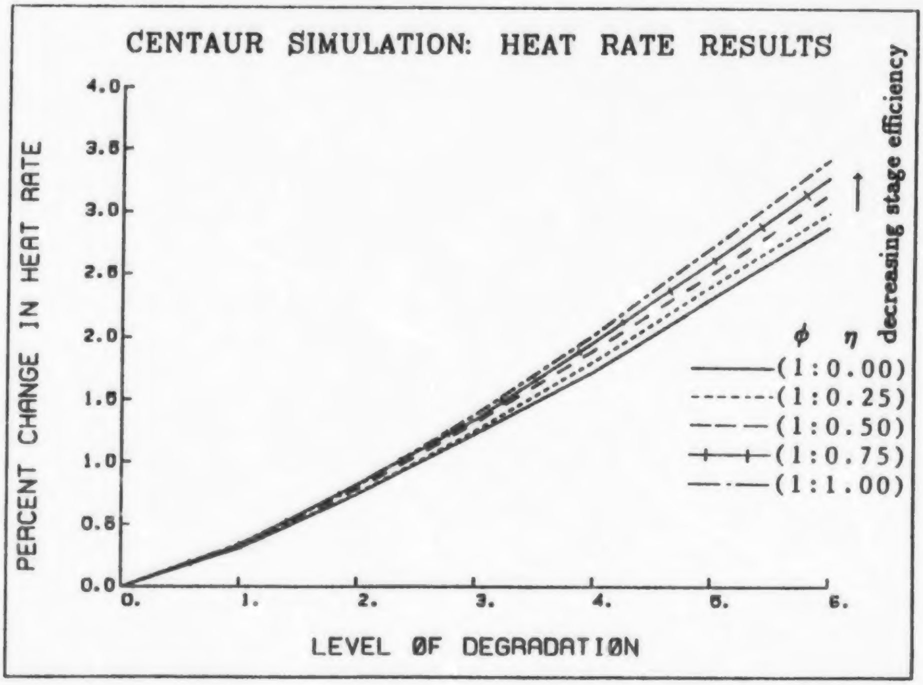


Figure 8

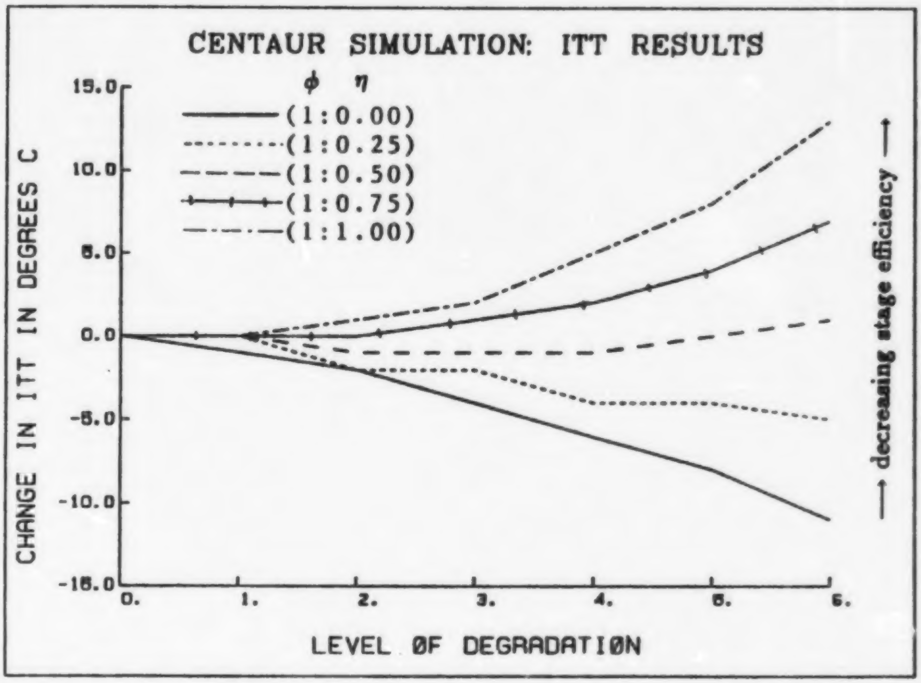

CENTAUR SIMULATION: ITT RESULTS
φ η
(1:0.00)
(1:0.25)
(1:0.50)
(1:0.75)
(1:1.00)
15.0
10.0
5.0
0.0
-5.0
-10.0
-15.0
CHANGE IN ITT IN DEGREES C
0.
1.
2.
3.
4.
5.
6.
LEVEL OF DEGRADATION
→ decreasing stage efficiency →

Figure 9

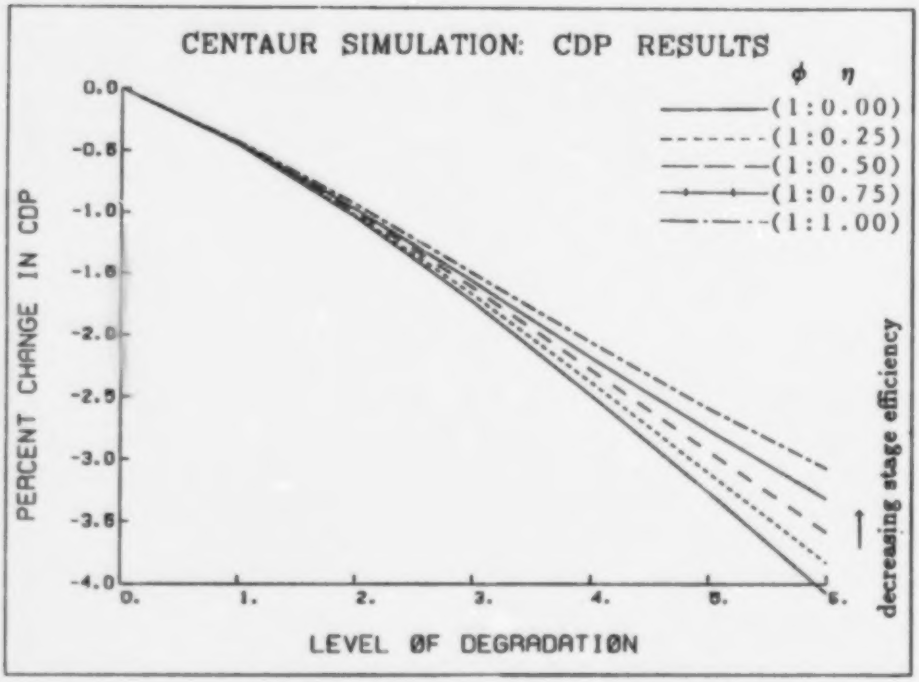


Figure 10

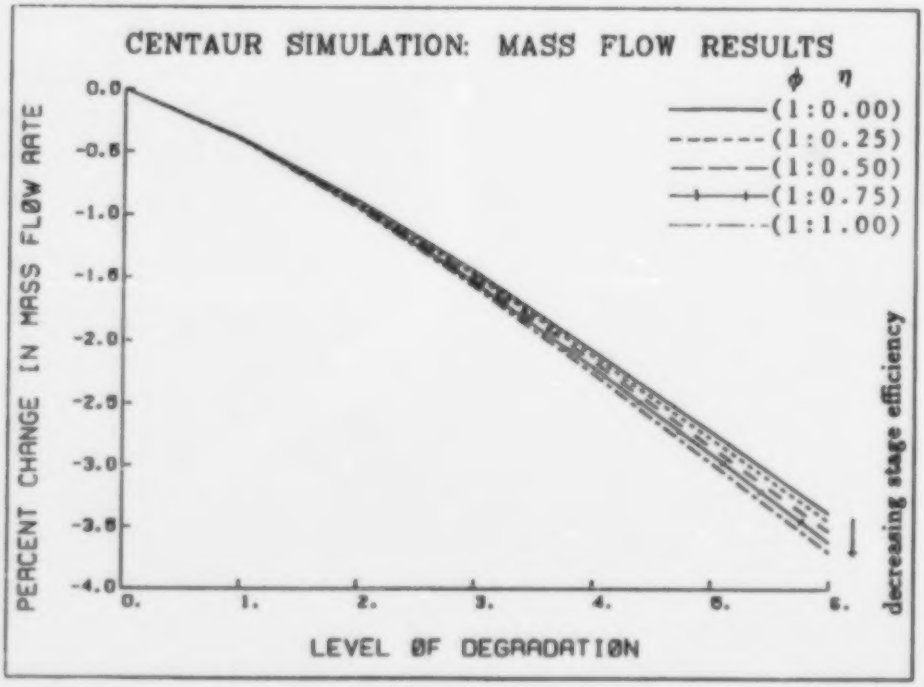


Figure 11

Paper prepared for presentation at the
Seventh NRCC Symposium on the Industrial
Application Of Gas Turbines to be held at
Banff Alberta 28 - 30 Sept 87

VIBRATION MONITORING EXPERIENCE IN GAS TURBINE ENGINES OF THE CANADIAN NAVY

Summary

Continuous monitoring of engine vibration proved to be difficult in ships of the DDH 280 Class and methods used to improve vibration measurement and analysis are described. Examples of useful results of vibration analysis are given and concepts of naval applications under current development are described.

AUTHOR - P. CHENEY P Eng C Eng MIMechE
DIRECTORATE OF MARINE AND ELECTRICAL ENGINEERING
NATIONAL DEFENCE HEADQUARTERS
OTTAWA

The opinions expressed in this paper represent those of the author and do not necessarily represent the policy of the Department of National Defence.

VIBRATION MONITORING EXPERIENCE IN GAS TURBINE ENGINES OF THE CANADIAN NAVY

In 1966 the Canadian Department of National Defence made the decision to introduce aircraft derived gas turbine engines into the forthcoming batch of Destroyer Class ships in place of the traditional steam power. This momentous decision was made all the more so by making the ships all gas turbine with a total of nine of these engines per vessel, with but a solitary diesel for standby electric power.

In the euphoria of entering the modern age of ship propulsion and with no comparable experience to draw upon, the importance and even the interest in health monitoring was minimized. After all, the engines were to be operated at only their "normal rating" which was defined as the maximum power at which service life would not be reduced by the power level alone. Thus a life expectancy lying between that of the gas pumper with it's benign environment and the aero application with its rapid power changes and high power take-offs was to be expected. Engine overhaul at four yearly ship refits or even longer periods seemed reasonable as planning criteria.

With these expectations it is not surprising that engine monitoring instruments feeding the central monitoring and control computers fitted on these ships were considered as health confirmation rather than diagnostic tools. Accuracies were set at that which was readily available rather than that which might be required, and display updates were a leisurely one minute. Current readings could be printed out at will by the operator seated in the central Machinery Control Room and an hourly review of the more important parameters was automatically logged. Another printer logged every change of speed ordered. Two parameters which were not fed to the computer were the fuel consumed and the vibration levels of the propulsion engines. Indeed neither of these parameters were even measured on the non-propulsion engines. Precision health monitoring was therefore off to a bad start with a shortage of accurate and reliable data being available.

This was the situation as the Navy entered the era of Reliability Centered or Condition Based Maintenance where engine health is the basis for repair rather than an arbitrary life. Put simply " if it ain't broke don't fix it ". To this end there has been much imaginative work into instrumentation and processing techniques aimed at determining the economic time at which to commence a maintenance or repair function. Investigation of oil analysis, gas path analysis, oil filter debris inspection and ferrography have shown that all of these have relevance to engine health monitoring but as health monitoring by vibration is the main theme of this paper I would like to delve a little deeper into the details of this technique as applied in DDH 280 Class ships.

Continuous measurement of propulsion engine vibration was originally accomplished using several piezo accelerometers feeding a remote signal conditioning unit fitted with a wide bandpass filter to register all rotating frequencies. The average vibration level was detected and a series of contact closures sent "warning" or "alarm/trip" messages to the engine control computer. Vibration levels to trigger these messages were based on the engine manufacturers' recommendations.

The automatic trip function was disconnected very early in the trials program of the ships owing to the propensity of the vibration measuring system to interpret electromagnetic interference and mechanical disturbance as dire levels of vibration, thereby tripping an engine, sometimes even before it started. As the display was similarly affected, use of the installed systems was discontinued and a fall-back measure was employed which used a portable vibration meter with manual selection of the fixed accelerometers. An attempt to replace this system with an improved installed system met with little enthusiasm although the prototype is still in use in one ship. Thus the first attempt to provide a fitted system to continuously monitor vibration on the ship propulsion engines was somewhat disappointing although many useful pointers were learned. Periodic monitoring using the manual systems has been of some value to the operators but as with the non-propulsion engines and other ship machinery, vibration readings have tended to be used to confirm satisfactory installation and maintenance work. When trouble has occurred or has been developing, vibration levels have rarely been recorded.

Following from this experience has been a recent effort to improve upon the specification and design of the fixed monitoring system. Although not yet installed, the trials of the resulting new equipment have been encouraging and shipboard installation is imminent. The basic specification differs little from that of the original equipment but many details have been improved. The piezo accelerometers selected use a sealed cable link to a local amplifier which boosts the signal power to a value which enables normal instrumentation cable to be run to the remote conditioning and readout equipment without fear of troublesome pickup. High output accelerometers with built-in electronics were considered for this application but their installation was precluded by the high temperature locations of some of the accelerometers. In the Machinery Control Room the electronic unit uses an LED bar graph to display the vibration of the point selected, averaged over the frequency range of 40 - 300 Hz on a scale registering in mils peak-to-peak. An important additional feature is that outputs are available of both the filtered and raw signals for recording or further analysis and for display by the Integrated Machinery Control System soon to be installed in these ships.

The potential to carry out deeper analysis and perhaps diagnosis was realized with the advent of the relatively low cost Fast Fourier Transform (FFT) analyzer which, given satisfactory signal inputs could provide very accurate discrete waveform analysis. Efforts were therefore directed at investigating the use of such an analyzer for naval applications. It was soon recognized that the originally fitted accelerometers were unsuitable as they produced a high level of spurious low frequency signals probably as a result of mechanical shock during handling. Also the extremely high impedance of the piezo devices coupled with very low power output gave a high sensitivity to electrical leakage and to cable damage, both difficult to avoid in the marine environment. Thus, trials of discrete vibration analyzers on propulsion engines aboard ship commenced with the temporary installation of "lab" accelerometers and wiring leading to the Machinery Control Room where the analyzer was installed together with tape recording equipment for further analysis ashore. To provide a practical means of displaying the large amount of data which is produced by discrete analysis, the data was presented in cascade format using axes of vibration frequency, rotor speed and vibration amplitude. Live engine testing comprised a slow acceleration from idle to maximum power over a period of about ten minutes followed by a similar deceleration. Due to practical limitations and to save testing time both main or both cruise engines were operated together, which although introducing the possibility of vibration linkage, was found to be a satisfactory procedure. Real-time analysis was restricted to time domain monitoring with occasional frequency displays to verify the correct functioning of the instrumentation. The cascade plots were produced later by shore based equipment by replaying the recorded data.

After some improvements in the format of the display the readability of the cascade plots was found to be excellent with discernable vibration at the fundamental and harmonic frequencies of each rotor in most instances. Data in this form was collected from each of the ships and from test bed runs on newly overhauled engines, to produce a data base of vibration levels which could be used as a basis line of equipment health as required for quality control and for fault diagnosis.

It soon became apparent that each individual engine had its own vibration signature probably related to the degree of balance and to the quality of the installation; the former giving rise to fundamental frequencies while the latter is known to provoke the higher harmonics. Another characteristic which was well demonstrated was resonance of engine components as the speed swept through individual natural frequencies.

An interesting case history was that of an engine which exhibited a higher than normal vibration in the mid speed range. Although not above the limits it was decided to remove the engine for investigation. Test bed running confirmed the high vibration level which appear fair quickly, peak and then diminish as speed was increased thus suggesting resonance. It was apparent from the cascade plots that excitation was from the low pressure spool and reference to the modal analysis and the measurement planes suggested that the problem was located near the forward turbine bearing pack. On disassembly it was found that a bearing in the turbine zone had been damaged, probably on initial assembly and that the roller cage was bent, causing some rollers to run askew. As a consequence the roller track had worn and caused increased radial clearance in the bearing. A review of the design of that part of the engine suggested that the bearing in question was probably designed to increase shaft rigidity to prevent whirl. It is thought that the vibration observed resulted from incipient shaft whirl which was present because of the reduced support from the worn bearing thus explaining the onset of resonance. The resulting repair almost certainly saved a more serious and costly failure.

In another case an engine was found to have developed a high vibration signature, again in the turbine zone. This time the amplitude was up to the limit and the engine was removed. strip inspection showed that the rear turbine bearing was out of alignment thus explaining the vibration. In this instance the original cause of the problem was traced to a seized front mount which allowed thermal strains to produce engine frame bending. Improved maintenance procedures were introduced following this incident.

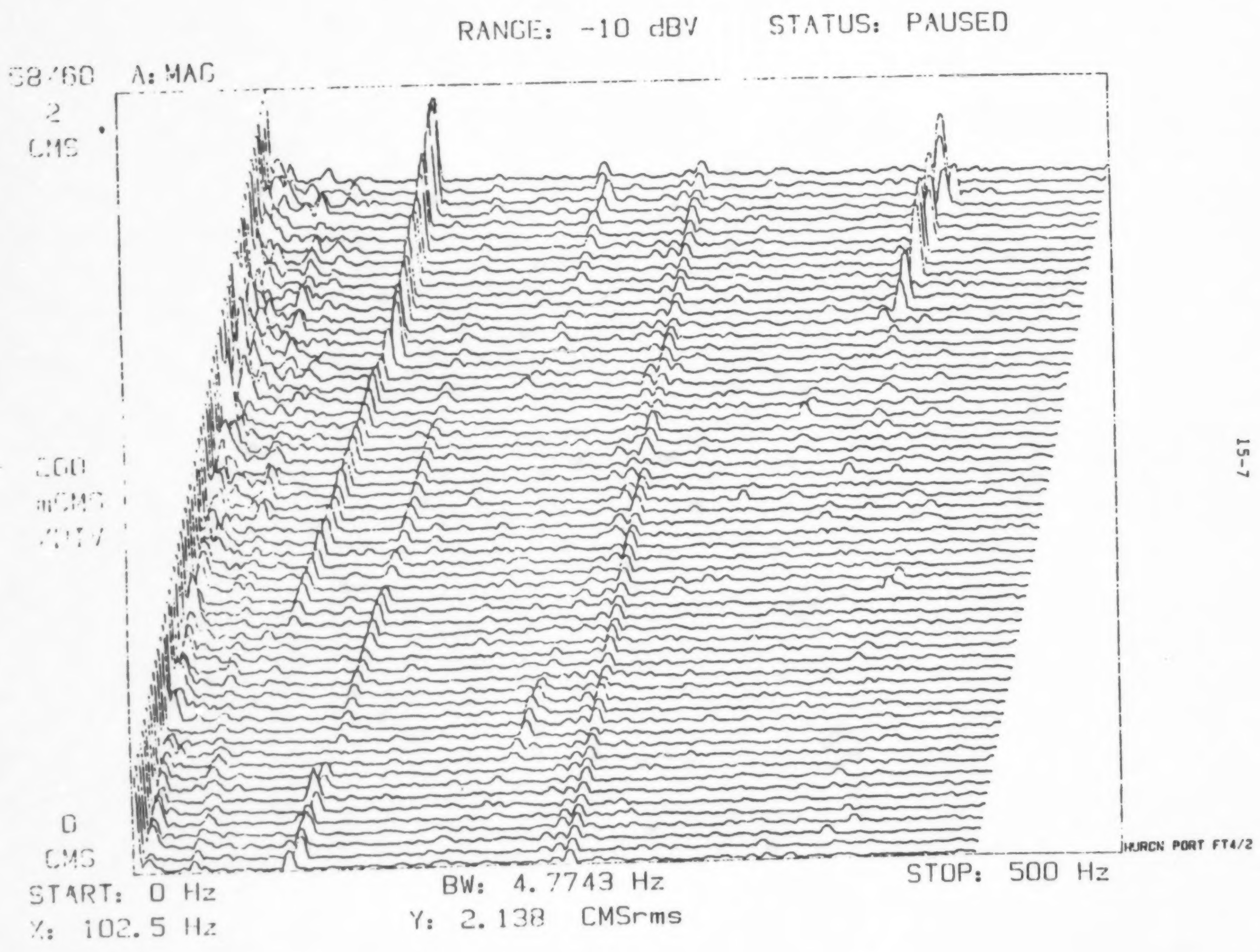
The long and expensive installation procedure required at present to prepare for a discrete vibration analysis run has reduced the frequency of the testing, but with the installation of the new fixed system and the accessibility of the new accelerometers it is planned to plot data on each engine routinely at annual full power trials.

Another avenue which is being investigated is the vibration level at blade passing frequencies. These occur at between ten and one hundred times the shaft rotation speeds and are usually present at high vibration velocities due to the aerodynamic loads and the light structure of the engines. Observations show a marked variation between otherwise normal engines in this frequency range. Current thoughts are that this is related to aerodynamic rather than mechanical causes and may well be a measure of the blade wake turbulence resulting from compressor load. This variation might derive from matching tolerances or from blade damage; in the latter case there may well exist balance anomalies resulting in shaft frequency vibrations thereby providing a diagnostic clue. An increasing trend in blade passing frequency vibration amplitudes is thought to be an indication of compressor fouling.

Yet another area of endeavor with which DND has been involved is an even more sophisticated approach to vibration monitoring. A system which makes use of phase triggers on each rotating element to focus attention on a particular element has been used successfully in the UK for analysis of gear box problems. A discrete analysis is made of those vibrations which are synchronized with the element thus enabling discrimination of low level vibrations in a high vibration environment. Pattern recognition techniques are then used to identify a particular known fault mode at an early stage. With experience a degree of prognosis should possible with economic benefits resulting from maximizing the useful life of engine components.

Testing of the equipment on an engine test bed showed that although placement of the necessary transducers was somewhat difficult the operating results were impressive in the hands of an expert analyst. The computing load of this system is large and therefore costly hardware is required, but with the ever reducing cost and size of computing elements it is considered worth while to pursue the system and software development and to acquire the data bases relative to the Navy engines with a view to future inclusion in the fitted suit of health monitoring equipment. In the meantime the equipment is seen as being very suitable for engine test house use as a sensitive analyzer of component vibration and as a means of providing the data bases.

As for the future, the DDH 280 Class, which is about to enter a major update program known generally as TRUMP, will have engine vibration levels continuously monitored and displayed in octave bands, with warnings of high levels and with sampling points for portable equipment as described. The new Canadian Patrol Frigates will have many machinery health monitoring capabilities including a combination of discrete vibration analysis and long term recording of vibration levels built into the Integrated Machinery Control Systems. Shore based facilities will continue to observe fleet machinery performance and investigate new methods of vibration measurement in this most promising facet of health monitoring. This work will be in cooperation with both the engine and instrument industries and with other government agencies. By this means a core of specialist expertise and equipment will remain available which in the short term may be employed to aid on-board diagnostic resources, or at least provide a second opinion, while in the long term the next generation of vibration analyzers will be defined to meet the ongoing demands of the fleet for ever greater engine availability.


RANGE: -10 dBV
STATUS: PAUSED
58/60
A: MAG
2
CMS
200
mCMS
/DIV
0
CMS
START: 0 Hz
X: 102.5 Hz
BW: 4.7743 Hz
Y: 2.138 CMSrms
STOP: 500 Hz
HURCN PORT FT4/2

APPENDIX

List of Attendees

Abbey, W.
Boiler/ Machinery Inspector
Royal Insurance
10 Wellington St. E.
Toronto, Ontario M5E 1L5

Abdelrazik, A.
Senior Project Engineer
Naval Engineering Test Est.
161 Wannklyn
Ville Lasalle, Quebec H8R 1Z2

Aker, G.F.
Engineer
TransCanada Pipelines
P.O. Box 54 Commerce Court West
Toronto, Ontario M5L 1C2

Alexander, C.B.
General Sales Manager
Solar Turbines Canada Ltd.
Suite 1720 Bow Valley Square 1
202 - 6th Avenue S.W.
P.O. Box 9474
Calgary, Alberta T2P 2R9

Alleyne, D.
Staff Machinery Engineer
Shell Canada Resources Ltd.
P.O. Box 100, Station M
Calgary, Alberta
T2P 2H5

Anderson, D.J
Supervisor, Technical Services
Westcoast Transmission Com.
1333 West Georgia Street
Vancouver, British Columbia V6E 3K

Anderson, P.S.
Manager
Pipeline and Facilities Engineering
NOVA, An Alberta Corp
P.O. Box 2535, Postal Station M
801 Seventh Avenue
Calgary, Alberta T2P 2N6

Armstrong, D.
Foreman
NOVA, An Alberta Corp
P.O. Box 2535, Postal Station M
801 Seventh Avenue
Calgary, Alberta T2P 2N6

Azarniouch, M.K.
Head of Energy Research Section
Paprican
570 St. John's Blvd.
Point Claire, Que. H9R 3J9

Bacle, J.F.
Sales Manager of Oil and Gas Industry
Industrial and Marine Division
Pratt & Whitney Canada Inc.
C.P. 10,
Longueuil, Quebec J4K 4X9

Barber, H.
Mechanical Engineering Specialist
Esso Resources Canada Ltd.
237 Fourth Avenue S.W.
Calgary, Alberta T2P 0H6

Barrie, J.G.
Mechanical and Piping Engineering
Fluor Canada Ltd.
10101 Southport Road ,S.W.
P.O. Box 8799, Station F
Calgary, Alberta T2W 3N2

Bonnor, I.
Superintendent
Saskatchewan Power Corp.
2025 Victoria Avenue
Regina, Saskatchewan S4P 0S1

Bowes, L.S.
Senior Mechanical Technician
Alberta Power Ltd.
10035 - 105 St.
Edmonton, Alta. T5J 2V6

Boyle, K.
Pipeline Engineer
Alberta Natural Gas Co. Ltd.
24th floor East Tower Esso Plaza
425 - 1st Street S.W.
Calgary, Alberta T2P 3L8

Brooks, R.
Product Support Technical Representative
Ruston Gas Turbines Ltd.
P.O. Box 1
Lincoln, England
LN2 5DJ

Bruce, D.H.
Support Techincal Services N.D.
Westcoast Transmiss
Bag Service 6180
First Street
John, British Columbia V1J 4H7

Campbell, A.M.
Sales Engineer
Dresser-Rand Canada Ltd.
206, 610 - 70th Ave.
Calgary, Alberta T2H 2J6

Campbell, I.G.
National Research Council Canada
Division of Mechanical Engineering
Gas Dynamics
Ottawa, Ontario K1A 0R6

Carlson, M.E.
Manager Operations
Foothills Pipelines (Yukon)
3000, 707 - 8 Ave. S.W.
Calgary, Alberta T2P 3W8

Chapman, T.
Manager Package Engineering
Solar Turbines Inc.
9250 Skypark Court
San Diego, California 92123

Cheney, P.
DMEE 2-2
National Defence
DMEEZ-2, 3 LSTL
NDHQ 101 Colonel By Drive
Ottawa, Ontario K1A 0K2

Christie, A.
Manager Gas Turbine Marketing
General Electric
Mail Zone N-158
Cincinnati, Ohio
45215

Colwell, P.J.
Compressor Station Engineer
Union Gas Limited
50 Keil Drive North
Chatham, Ontario N7M 5M1

Conquergood, C.P.
Head Engineer - Machinery
Shell Canada Limited
400 - 4th Ave. S.W.
Calgary, Alberta T2P 2H5

Coulter, D.
Gas Turbine Training Officer
CFFS Halifax
FMO Halifax
Nova Scotia, B3K 2X0

Crockford, R.W.
Operations Superintendent
City of Medicine Hat
580 - 1st Street S.E.
Medicine Hat, Alberta T1A 8E6

Cumberland, R.P.
Supervising Engineer, Facilities Engineering
NOVA, An Alberta Corp
P.O. Box 2535, Postal Station M
801 Seventh Avenue
Calgary, Alberta T2P 2N6

Dallas, M.
Operations Foreman
NOVA, An Alberta Corp.
P.O. Box 670
Rocky Mountain House
Alberta T0M 1T0

Dawson, J.
Manager, Propulsion Systems
Computing Devices Co.
P.O. Box 8508
Ottawa, Ontario K1G 3M9

Diakunchuk, I.S.
Manager
Westinghouse Canada Ltd.
P.O. Box 510
Hamilton, Ontario L8N 3K2

Donnelly, M.
Western Canada Sales Manager
Flexibox (Canada) Inc.
P.O. Box 368
Nisku, Alberta T0C 2G0

Downton, S.
Technologist
NOVA, An Alberta Corp
P.O. Box 2330
9888 Jasper Avenue
Edmonton, Alberta T5J 2R1

Ducholke, R.
Senior Maintenance Plan Technician
NOVA, An Alberta Corp
P.O. Box 2330
9888 Jasper Avenue
Edmonton, Alberta T5J 2R1

Dyke, L.C.
Senior Staff Engineer
Standard Aero Limited
33 Allen Dyne Road
Winnipeg, Manitoba R3H 1A1

Eatock, H.C.
Asst. Director,
Analyical Engineering-Aerodynamics
Pratt and Whitney Canada
1000 Marie-Victorin
Longueuil, Quebec J4G 1A1

Evans, R.E.
Regional Manager
Cooper Industries (Canada) Inc.
870 Gulf Canada Square
401 - 9th Ave. S.W.
Calgary, Alberta T2P 3C5

Feldmeyer, C.R.
President
Alberfeld Enterprises Inc.
426 Oakhill Place S.W.
Calgary, Alberta T2V 2X6

Fertuck, E.A.
Project Sales
AAF Canada
525 Hodge Street
Montreal, Que. H4N 2A3

Fournier, J.-M.
Sales Manager Electrical Utilities
Industrial and Marine Division
Pratt & Whitney Canada Inc.
C.P.10,
Longueuil, Quebec J4K 4X9

Germaine, K.
Sales Engineer
Bently Nevada Canada Ltd.
P.O. Box 233
Nisku, Alberta T0C 2G0

Glew, W.
Special Projects Engineer
Naval Engineering Test Est.
161 Wanklyn
Ville LaSalle, Quebec H8R 1Z2

Gorski, J.P.
Inspector
The Boiler Inspection and
Insurance Company of Canada
147 Penmeadows Cl., S.E.
Calgary, Alberta T2A 3S1

Gough, G.
Nuclear Technican
Ontario Hydro
P.O. Box 160
Pickering, Ontario L1V 2R5

Grav, M.
Rotating Equipment Engineer
Lagoven SA
P.O. Box 172
Maracaibo 4001, Venezuela

Handy, P.J.
Engineering Sales Manager
Rolls-Royce (Canada) Ltd.
Industrial & Marine Division
P.O. Box 544, Montreal AMF
Montreal, Quebec H4Y 1B3

Hartley, D.
Senior Operator
NOVA, An Alberta Corp.
P.O. Box 670
Rocky Mountain House
Alberta T0M 1T0

Henry, C.A.
Senior Technologist
Chevron Canada Resources Ltd.
500 Fifth Avenue S.W.
Calgary, Alberta T2P 0L7

Heuchert, D.E.
Supervising Engineer, Facilities Engineering
NOVA, An Alberta Corp
P.O. Box 2535, Postal Station M
801 Seventh Avenue
Calgary, Alberta T2P 2N6

Hobbs, K.C.
Head of Product Support (Industrial and Marine)
Rolls-Royce (Canada) Ltd.
Industrial & Marine Division
P.O. Box 544, Montreal AMF
Montreal, Quebec H4Y 1B3

Hockey, S.C.
Inspector
Gerling, Global Gen Ins.
4271 Louisburg Pl.
Richmond, British Columbia

Hreczuch, W.
Senior Mechnical Engineer
Alberta Power Ltd.
10035 - 105 St.
Edmonton, Alta. T5J 2V6

Hubble, D.C.
Technical Support
Ontario Hydro
700 University Avenue
Toronto, Ontario M5G 1X6

Immarigeon, J.P.
Deputy Laboratory Head
National Aeronautical Est.
National Research Council Canada
Montreal Rd., Bldg. M13
Ottawa, Ontario K1A 0R6

Inglis, J.P.
Vice-President, Product Support
Rolls-Royce (Canada) Ltd.
Industrial & Marine Division
P.O. Box 544, Montreal AMF
Montreal, Quebec H4Y 1B3

Johnson, D.
Technologist
NOVA, An Alberta Corp
P.O. Box 2330
9888 Jasper Avenue
Edmonton, Alberta T5J 2R1

Johnston, C.
Supervisor Operations Engineering
NOVA, An Alberta Corp
P.O. Box 2330
888 Jasper Avenue
Edmonton, Alberta T5J 2R1

Junkey, R.C.
Sales
Cooper Energy Services Ltd.
6889 Rexwood Rd.
Mississauga, Ont. L4V 1R2

Junkin, B.
Manager Engineering Technical Services
Standard Aero Ltd.
33 Allen Dyne Road
Winnipeg, Manitoba R3H 1A1

Kalman, M.
Controls Foreman
NOVA Corporation of Alberta
Box 1650
Vegreville, Alberta T0B 4L0

Kardash, R.J.
Regional Technician
TransCanada PipeLines
1801 Hamilton Street, Suite 1120
Regina, Saskatchewan S4P 4B4

Karim, G.A.
Professor
University of Calgary
Department of Mechanical Engineering
500 University Dr. N.W.
Calgary, Alberta T2N 1N4

King, A.
Senior Engineer
British Columbia Hydro
4-1265 Howe Street
Vancouver, British Columbia
V6Z 2C8

Knight, B.K.
Vice-President
Rolls-Royce (Canada) Ltd.
Industrial & Marine Division
P.O. Box 544, Montreal AMF
Montreal, Quebec H4Y 1B3

Kraft, H.
Pipeline Engineer
Alberta Natural Gas Company
P.O. Box 39
Cranbrook, British Columbia V1C 4H6

Kuczynski, M.
Product Support Engineer
Solar Turbines Canada Ltd.
2510-84th Avenue
P.O. Box 5630, Station L
Edmonton, Alberta T6C 4G1

Laws, D.V.
Chief Sales Engineer
Ruston Gas Turbines Ltd.
P.O. Box 1
Lincoln, England
LN2 5DJ

Little, D.A.
Senior Engineer
Liburdi Engineering Ltd.
400 Highway 6 North
Hamilton, Ontario L9J 1E7

McCachen, W.G.
Director Canadian Sales
Dresser-Rand Canada
206, 610-70th Ave.SE.
Calgary, Alta. T2H 2J6

McEnhill, J.A.
Reed Stenhouse Ltd.
900 Howe St.
Vancouver, B.C. V6B 3X6

McLarty, V.
Staff Engineer
Standard Aero Limited
33 Allen Dyne Road
Winnipeg, Manitoba R3H 1A1

Meier, R.H.
Cooper Industries
North Sandusky Street
Mt. Vernon, Ohio 43050

Morland, R.
Maintenance Technician
Maintenance Technician
Esso Resources Canada Ltd.
Mail Bag 1000
Swan Hill, Alberta T0G 2C0

Moses, P.B.
Project Manager
Syncrude Canada Ltd.
c/o Bantrel Group, Brittania Bldg.
8th Floor 703 - 6th Ave. S.W.
Calgary, Alberta T2P 4C4

Nicholls-Gossey, L.
Operations Manager
CSR Oil and Gas Position in
Queensland Australia
Associated Pipelines Ltd.
G.P.O. Box 880
Brisbane, Queensland
Australia 4001

Nelson, W.D.
Marketing and Sales
HSDE Inc.
2869 Higgens Road
Elk Grove, Illinois 60007

Nevin, D.R.
Engineer
Westinghouse Canada Inc.
P.O. Box 510,
Hamilton, Ontario L8N 3K2

Nicholls-Gossey, L.
Operations Manager
CSR Oil and Gas Position in
Queensland Australia
Associated Pipelines Ltd.
G.P.O. Box 880
Brisbane, Queensland
Australia 4001

Niven, M.J.
District Service Manager
Dresser Rand Canada Inc.
206, 610 - 70th Ave. S.E.
Calgary, Alberta T2H 2J6

Parker, R.
Project Engineer, Operation Engineering
NOVA, An Alberta Corp
P.O. Box 2330
9888 Jasper Avenue
Edmonton, Alberta T5J 2R1

Patadia, P.
Delta Projects Inc.
P.O. Box 5244, Station A
Calgary, Alberta T2H 1N7

Penney, W.R.
Assistant Supervising Engineer
Compression Facilities
TransCanada Pipelines
P.P. Box 54
Commerce Court West
Toronto, Ontario M5L 1C2

Ragland, T.
Manager
Analytical and Performance Engineering
Solar Turbines Inc.
2200 Pacific Hwy., P.O. Box 85376
San Diego, California 92138-5376

Randall, E.H.
Engineering Supervisor
Foothills Pipelines
3000 , 707 - 8 Ave. S.W.
Calgary, Alberta T2P 3W8

Rogers, G.
Project Manager
Dresser-Rand
6688 Kitimat Road
Mississauga, Ontario L5N 1P8

Sampath, S.P.
Chief of Combustion Engineering
Pratt & Whitney Canbada Inc.
P.O. Box 4080
Mississauga, Ontario L5A 3Z4

Saravanamuttoo, H.I.H.
Chairman, Professor
Carleton University
Dept. Mechanical &
Aeronautical Engineering
Ottawa, Ontario K1S 5B6

Sawbridge, B.E.
Project Engineer
Pratt & Whitney Canada Inc.
1000 De Serigny Street
Longueuil, Quebec J4K 5B1

Sawyer, G.H.
Marketing Director
Vibro-Meter Inc.
2405 de la Province
Longueuil, Quebec J4G 1G3

Scott, J.
John Scott Engineering
6731 Lachine Court S.W.
Calgary, Alberta T3E 6E7

Scott, K.
Nuclear Technician
Ontario Hydro
Bruce N.G.S. 'B' Operations
Box 4000
Tiverton, Ontario N0G 2T0

Smith, G.E.
Systems Engineer
Cooper Industries
Entronics
North Norton Street
Mt. Vernon, Ohio 43050

Stitt, D.H.
Engineer
Pacific Gas Transmission Comp.
160 Spear Street
San Francisco, California

Stoykewich, S.G.
Manager, Repair and Overhaul
Vac-Aero International Inc
1371 Speers Road,
Oakville, Ontario L6L 2X5

Swatton, D.B.
Senior Site Installation Rep.
Pratt & Whitney Canada Inc.
1000 De Serigny Street
Longueuil, Quebec J4K 5B1

Tadros, F.
Staff Applications Engineer
Pratt & Whitney Canada Inc.
1000 De Serigny Street
Longueuil, Quebec J4K 5B1

Tessier, L.P.
Machinery Engineering Specialist
Esso Resources Canada Ltd.
Esso Plaza West
237 - 4th Avenue, S.W.
Calgary, Alberta T2P 0H6

Tuckett, R.C.
Senior Technician
OACETT
TransCanada Pipelines
P.O. Box 54, Commerce Court W.
Toronto, Ontario M5L 1C2

Turley, M.R.
Customer Services Representative
Solar Turbines Canada Ltd.
Suite 1720, Bow Valley Square
202- 6th Avenue S.W.
P.O. Box 9474
Calgary, Alberta T2P 2R9

Van Hardeveld, T.
Maintenance Engineering Specialist
NOVA, An Alberta Corp
P.O. Box 2330
9888 Jasper Avenue
Edmonton, Alberta T5J 2R1

Walls, K.G.
Senior Sales Engineer
Solar Turbines Canada Ltd.
Suite 1720 Bow Valley Square 1
202 - 6th Avenue S.W.
P.O. Box 9474
Calgary, Alberta T2P 2R9

West, B.
Engineering Specialist
Royal Insurance
10 Wellington St. E.
Toronto, Ontario M5E 1L5

Whynot, R.
Supervising Inspector
Royal Insurance
10 Wellington St. E.
Toronto, Ontario M5E 1L5

Wierzba, I.
Professor
University of Calgary
Department of Mechanical Engineering
2500 University Dr. N.W.
Calgary, Alberta T2N 1N4

Williams, L.J.
Supervising Engineer
TransCanada Pipelines
P.O. Box 54 Commerce Court West
Toronto, Ontario M5L 1C2

Williamson, R.G.
Assistant Director
Head, Gas Dynamics Laboratory
Division of Mechanical Engineering
National Research Council Canada
Ottawa, Ontario K1A 0R6